U.S. DEPARTMENT OF COMMERCE
National Technical Information Service

PB-276 154

Current Status of Alternative Automotive Power Systems and Fuels. Volume II Alternative Automotive Engines

Aerospace Corp, El Segundo, Calif

Prepared for

Environmental Protection Agency, Ann Arbor, Mich

Jul 74

PB 276 154

EPA-460/3-74-013-b
July 1974

CURRENT STATUS OF ALTERNATIVE AUTOMOTIVE POWER SYSTEMS AND FUELS

VOLUME II - ALTERNATIVE AUTOMOTIVE ENGINES

REPRODUCED BY
NATIONAL TECHNICAL
INFORMATION SERVICE
U. S. DEPARTMENT OF COMMERCE
SPRINGFIELD, VA. 22161

U.S. ENVIRONMENTAL PROTECTION AGENCY
Office of Air and Waste Management
Office of Mobile Source Air Pollution Control
Alternative Automotive Power Systems Division
Ann Arbor, Michigan 48105

TECHNICAL REPORT DATA

(Please read Instructions on the reverse before completing)

1. REPORT NO. EPA-460/3-74-013-b	2.	3. RECIPIENT'S ACCESSION NO. PB276154
4. TITLE AND SUBTITLE Current Status of Alternative Automotive Power Systems and Fuels Volume II - Alternative Automotive Engines		5. REPORT DATE July 1974 6. PERFORMING ORGANIZATION CODE
7. AUTHOR(S) D. E. Lapedes, M. G. Hinton, J. Meltzer, T. Iura, O. Dykema, L. Forrest, F. Ghahremani, R. Kopa, W. Lee, W. Roessler, W. Smalley		8. PERFORMING ORGANIZATION REPORT NO. ATR-74(7325)-1, Vol. II
9. PERFORMING ORGANIZATION NAME AND ADDRESS The Environmental Programs Group Environment and Urban Division The Aerospace Corporation El Segundo, California 90245		10. PROGRAM ELEMENT NO. 11. CONTRACT/GRANT NO. 68-01-0417
12. SPONSORING AGENCY NAME AND ADDRESS EPA, Office of Air and Waste Management Office of Mobile Source Air Pollution Control Alternative Automotive Power Systems Division Ann Arbor, Michigan 48105		13. TYPE OF REPORT AND PERIOD COVERED Final 14. SPONSORING AGENCY CODE

15. SUPPLEMENTARY NOTES

16. ABSTRACT

A summarization has been made of the available nonproprietary information on the technological status of automotive power systems which are alternatives to the conventional internal combustion engine, and the technological status of non-petroleum-based fuels derived from domestic sources which may have application to future automotive vehicles. The material presented is based principally upon the results of research and technology activities sponsored under the Alternative Automotive Power Systems Program which was originated in 1970. Supplementary data are included from programs sponsored by other government agencies and by private industry. The results of the study are presented in four volumes; this volume presents available information pertaining to advanced alternative automotive heat engines.

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS	b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
Automotive Heat Engines Gas Turbine Rankine Cycle Stirling Cycle Diesel Wankel (Rotary Piston) Stratified Charge	Exhaust Emissions Fuel Economy Technology Status	
18. DISTRIBUTION STATEMENT Unlimited	19. SECURITY CLASS *(This Report)* Unclassified	21. NO. OF PAGES 463
	20. SECURITY CLASS *(This page)* Unclassified	22. PRICE PC A20-A01

EPA Form 2220-1 (9-73)

EPA-460/3-74-013-b

CURRENT STATUS OF ALTERNATIVE AUTOMOTIVE POWER SYSTEMS AND FUELS

VOLUME II - ALTERNATIVE AUTOMOTIVE ENGINES

Prepared by

The Environmental Programs Group

The Aerospace Corporation
El Segundo, California 90245

Contract No. 68-01-0417

EPA Project Officer: Graham Hagey

Prepared for

U.S. ENVIRONMENTAL PROTECTION AGENCY
Office of Air and Waste Management
Office of Mobile Source Air Pollution Control
Alternative Automotive Power Systems Division
Ann Arbor, Michigan 48105

July 1974

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This report was furnished to the U. S. Environmental Protection Agency by the Aerospace Corporation, El Segundo, California, in fulfillment of Contract No. 68-01-0417 and has been reviewed and approved for publication by the Environmental Protection Agency. Approval does not signify that the contents necessarily reflect the views and policies of the agency. The material presented in this report may be based on an extrapolation of the "State-of-the-art". Each assumption must be carefully analyzed by the reader to assure that it is acceptable for his purpose. Results and conclusions should be viewed correspondingly. Mention of trade names or commercial products does not constitute endorsement or recommendation for use.

Publication No. EPA-460/3-74-013-b

FOREWORD

This report, prepared by The Aerospace Corporation for the Environmental Protection Agency (EPA), Alternative Automotive Power Systems Division, summarizes available nonproprietary information on the technological status of automotive power systems which are alternatives to the conventional internal combustion engine, and the technological status of nonpetroleum-based fuels derived from domestic sources which may have application to future automotive vehicles.

The status of the technology reported herein is that existing at the end of 1973 with more recent data in selected areas. The material presented is based principally upon the results of research and technology activities sponsored under the Alternative Automotive Power Systems (AAPS) Program which was originated in 1970 and which is administered by the Alternative Automotive Power Systems Division of EPA. Supplementary data are included from programs sponsored by other government agencies and by private industry. Additional information on technology and development programs is known to the government but cannot be documented herein because the data are proprietary.

One purpose that the AAPS Program serves is to provide a basis of knowledge and perspective on what can and cannot be accomplished with the use of alternative propulsion and fuels technology and to disseminate this information to Congress, Federal policy makers, industry, and the public. Thus, the publication of information such as that contained herein is in keeping with this element of the mission of the AAPS Program. This is the first of a series of reports on alternatives that are intended to be published annually.

The results of this study are presented in four volumes and three main topical areas:

Volume I. Executive Summary
Volume II. Alternative Automotive Engines

Volume III. Alternative Nonpetroleum-based Automotive Fuels
Volume IV. Electric and Hybrid Power Systems

Volume I, the Executive Summary, presents a concise review of important findings and conclusions for all three topical areas. Thus, an overview of the study results may be obtained by reading Volume I only. Volumes II, III, and IV contain detailed, comprehensive discussions of each topical area and are therefore of interest primarily to the technical specialist. Each of these three volumes also contains Highlights and Summary sections pertaining to the topical area covered in the volume.

This volume, Volume II, presents available information pertaining to the current technological status of advanced alternative automotive heat engines which may have application to future automotive vehicles.

A brief review of important findings and conclusions is presented in the Highlights and Summary sections. A short discussion of the background of the Alternative Automotive Power Systems (AAPS) Program sponsored by EPA is given in Section 1, Introduction. Section 2 presents a brief historical review of the evolution of the spark ignition engine powered automobile and discusses automobile systems and operational requirements with respect to the interrelationships influencing performance capability, fuel economy, and exhaust emissions as characterized by current Federal exhaust emissions standards. Sections 3 through 8 address in detail the available data concerning the six alternative engine classes examined: gas turbines, Rankine cycle engines, Stirling engines, diesel engines, Wankel and other rotary piston engines, and stratified charge engines. The principal topics covered for each engine class include power plant description, performance characteristics, and current and projected technology status. Appendices A and B contain a listing of vehicle specifications for full-size and compact passenger cars. These specifications were developed in the AAPS Program to provide a reference basis for comparison of vehicle performance when powered by various alternative engines.

ACKNOWLEDGMENTS

Appreciation is acknowledged for the guidance and assistance provided by Mr. Graham Hagey of the Environmental Protection Agency, Alternative Automotive Power Systems (AAPS) Division, who served as EPA Project Officer for this study. Appreciation is also extended to staff members in the AAPS Division, to other EPA divisions, to various government agencies and to those in industry and the academic community who supplied reference material and reviewed the contents of this report.

The following technical personnel of The Aerospace Corporation made valuable contributions to the effort performed under this contract:

Owen Dykema
Lester Forrest
Farhad Ghahremani
Richard Kopa
Warner Lee
Wolfgang Roessler
William Smalley

Merrill Hinton
Merrill G. Hinton, Director
Office of Mobile Source Pollution

DE Lapedes
D. E. Lapedes, Study Manager

Toru Iura
Toru Iura, Associate Group Director
Environmental Programs Group
Directorate

Joseph Meltzer
Joseph Meltzer, Group Director
Environmental Programs Group
Directorate

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HIGHLIGHTS

Numerous power system concepts have been proposed for road vehicles. In the category of heat engines for automobiles, several different types of engines have demonstrated the ability to provide the desired power levels while simultaneously offering the potential for major reductions in exhaust emissions when contrasted with the conventional spark ignition engine. In some cases, an engine has been characterized as having the potential to meet the original 1975 Federal emission standards (hydrocarbon = 0.41 grams per mile (gm/mi), carbon monoxide = 3.4 gm/mi, nitrogen oxide = 3.1 gm/mi) or the original 1976 standards (hydrocarbon = 0.41 gm/mi, carbon monoxide = 3.4 gm/mi, nitrogen oxide = 0.4 gm/mi) without extensive aftertreatment or add-on devices. Each engine, however, has certain design and performance deficiencies to be overcome (viz., excessive size, high manufacturing cost, prolonged startup time, etc.). Both industry and government have sponsored programs for overcoming these deficiencies through a combined application of analytical and test efforts. Recent concern with conservation of natural resources has prompted additional effort in reducing engine fuel consumption and verifying flexibility of engine operation using a wide range of possible fuels. This effort is exemplified by modifications within EPA's Alternative Automotive Power Systems (AAPS) Program to stress reducing fuel consumption as well as achieving emission levels consistent with Federal standards.

The following highlights briefly summarize the status of development work on alternative engines investigated in recent years. The engine types under review are: gas turbine, Rankine cycle, Stirling, diesel, Wankel (and other rotary piston designs), and stratified charge. These engine types are in various stages of development, ranging from redesign of engines under limited production to extensive research and development efforts on engines at least eight to nine years from full production (if placed in a production program status).

GAS TURBINE ENGINES

1. Until recently, the exhaust emissions of automotive-type gas turbine engines have been typified as characteristically low in unburned hydrocarbons and carbon monoxide, but with oxides of nitrogen levels too high to permit attainment of the original 1976 Federal emission standard of 0.4 gm/mi. For example, Chrysler's sixth-generation engine installed in an intermediate-size vehicle has NO_x emissions of about 2.2 gm/mi over the Federal Emissions Test Driving Cycle.

2. However, within the AAPS Program, Solar, a Division of International Harvester, has recently developed a new gas turbine combustion system that, based on steady-state test data, is calculated to be capable of meeting the original 1976 Federal emission standards and is close to meeting the AAPS Program goals (one-half the level of the original standards). Additional development is required to perfect the combustor control system and to validate meeting program goals with transient operation of the engine over the Federal Emissions Test Driving Cycle.

3. Reflecting similar recent advancements, a General Motors-sponsored program resulted in an experimental 225-horsepower gas turbine engine installed in a car meeting the original 1976 Federal emission standards. Complete details concerning engine design and operation are not available.

4. Estimated high production costs continue to be a major factor inhibiting the implementation of the automotive gas turbine engine. Other problems involve the need for improvements in noise level, durability, acceleration lag, and fuel economy. These problems are being investigated by Chrysler under a $6.5 million EPA contract awarded in December 1972 for the development of an improved version of its sixth-generation engine for powering an intermediate-size car. (Engine design for a compact car will be examined in a recent extension to this program.) The NASA Lewis Research Center is playing an important role in the effort in the areas of combustion, rotating machinery design, and engine test.

5. While demonstrated fuel economy is low for gas turbine engine-powered cars (about eight to nine miles per gallon for an intermediate-size car over the Federal Emissions Test Driving Cycle), it is estimated that fuel economy for improved engines will rise to about 12 miles per gallon and be competitive with that of current spark ignition engine-powered cars.

6. In the future, if ceramic turbines are developed, a substantial rise in turbine operating temperature will be possible, leading to a significant reduction in fuel consumption. For example, based on theoretical estimates, a temperature increase at the turbine inlet from a current

value of about 1800°F to a future value of 2500°F would result in about a 20 percent improvement in fuel consumption. An additional benefit might be reduced manufacturing cost.

RANKINE CYCLE ENGINES

1. Under the AAPS Program, the Rankine cycle engine is in an advanced state of development, having progressed to the point where several complete engine systems have been tested on engine dynamometers. Some of these engine systems are water-base working fluid designs (steam engine) that present the problem of fluid freezing at low temperatures. Others rely on an organic-type working fluid as one means of circumventing this problem.

2. Preliminary analyses of steady-state data from the AAPS Program show that the emissions from this engine may meet the original 1976 Federal emission standards and that fuel economy can be competitive with current emission-controlled spark ignition engines. A major breakthrough in design of the engine condenser has resulted in much smaller units, which has led to improved engine installation in current automobile engine compartments.

3. The task of repeated demonstrations of low emissions in transient operation (including cold start) with a fully automatic control system has been initiated under the AAPS Program. Scientific Energy Systems Corporation has been selected from several competing contractors to provide EPA with a vehicle-installed steam engine system for testing. This engine design utilizes a reciprocating piston expander for transmitting power through a transmission to the rear wheels (in contrast to other designs based on a turbine expander).

4. Future AAPS Program efforts are expected to concentrate on achieving improvements in fuel economy and investigating low-cost production techniques.

5. In addition to the EPA AAPS Program, the U.S. Department of Transportation has investigated Rankine cycle power systems for buses; and the State of California, under the Clean Car Project, has been testing this type of engine for passenger car use. Other domestic and foreign programs are being funded by private capital.

STIRLING ENGINES

1. The Stirling engine has the potential for demonstrating excellent fuel economy, multifuel capability, very low noise and vibration, and emissions low enough to meet the 1976 Federal standards.

2. The most significant advancement in recent times is a major reduction in engine volume, coupled with a major increase in power output per pound of engine weight. This has led to the initiation of a Ford demonstration program that involves the installation of engines in Torino and Pinto automobiles. N. V. Philips Laboratories at Eindhoven, Holland, will furnish the engine for the Torino installation, and United Stirling of Sweden will provide the Pinto engine.

3. Primary problem areas requiring further development work for resolution are:

   a. For efficient operation of this engine, the radiator in certain designs can be about two and one-half times as large as that for a comparable internal combustion engine. A reduction in radiator size will be necessary to permit satisfactory installation in current automobile engine compartments.

   b. To contain the high-pressure hydrogen working fluid, the array of heater tubes is now made of nickel-chrome alloys. These tubes are expensive to fabricate and, therefore, production processes must be improved or new designs must evolve to aid in reducing engine manufacturing costs.

4. Other problems with the Stirling engine include inadequate piston seal life, hydrogen diffusion through the cylinder walls and seals, and the need for low-cost power control capable of providing smooth power transitions. Engineering solutions are available, but they require further development and demonstration.

DIESEL ENGINES

1. A number of manufacturers (e.g., Mercedes, Peugeot, Opel) are currently marketing a passenger car powered by a diesel engine.

2. Based on test data taken by EPA, research institutions, and engineering/manufacturing firms, light-duty vehicles powered by diesel engines have emissions below the original 1975 Federal emission standards, but there are no indications as yet that this engine can adequately meet the original 1976 Federal NO_x standard. Additionally, diesel odor and particulates (smoke) have long been recognized as very undesirable exhaust emission products. Progress toward determination of the cause of the odor has been very slow, though its control has been achieved in some engines through fuel injection system refinements.

3. The fuel economy of diesel-powered vehicles over the Federal Emissions Test Driving Cycle is between 50 percent and 70 percent better than that achieved by the average 1973 model year emission certification vehicles tested at the same inertia weight. Currently, however,

because installed diesel engine power is relatively low, vehicle acceleration is significantly lower than that found with spark ignition engine-powered cars. On an equivalent performance basis, the fuel economy advantage of a higher-powered diesel vehicle would be expected to be smaller than that for current lower-powered diesel vehicles.

4. Power output per pound of engine weight is considerably lower than that presently offered in spark ignition engine-powered domestic automobiles. Methods for increasing the power output per pound of engine weight include: a) means to increase power by turbocharging or supercharging, and b) means to reduce engine weight and bulk by a reduction in engine compression ratio and a shortened design life.

WANKEL AND OTHER ROTARY PISTON ENGINES

1. Rotary engine-powered car production levels have been rising, although they represent a very small fraction of the automobile population. Currently, these engines appear principally in Mazda vehicles manufactured by Toyo Kogyo. Production levels are expected to increase with the limited introduction scheduled for the 1975 model year of the Chevrolet Vega powered by the General Motors rotary engine.

2. For a given power rating, the rotary engine is lighter and smaller than a reciprocating piston spark ignition engine. Reductions in engine size and weight can also lead to smaller and lighter vehicles as the result of more compact packaging.

3. A manufacturing problem exists in attempts to increase production rates. Because of complex machining operations required on both the housing and rotor, production of this engine is currently limited to about 25 per hour per line (compared to piston engine rates of over 100 per hour per line). This limitation appears to have a significant impact on manufacturing costs.

4. Technological problems still exist for some engine designs. These consist of high gas-seal leakage, high thermal stresses, poor low-speed fuel economy, and high exhaust emissions without aftertreatment. With aftertreatment (a thermal reactor), the Mazda rotary engine has demonstrated the ability to meet the original 1975 Federal emission standards.

STRATIFIED CHARGE ENGINES

1. Honda has a divided-chamber version (designated CVCC) of the stratified charge engine in production. Without incorporation of additional

emission control systems, its CVCC engine-powered Civic vehicle met the original 1976 Federal emission standards for HC and CO, while NO_x was about twice the standard. Vega and Impala vehicles achieved similar emission levels at low mileage when powered by General Motors engines modified by Honda to the CVCC configuration.

2. Army M-151 vehicles equipped with Texaco's and Ford's experimental open-chamber version of the stratified charge engine could meet the original 1976 Federal standards with the use of an emission control system incorporating oxidation catalysts and exhaust gas recirculation. But these vehicles were unable to negotiate the high acceleration modes of the Federal Emission Test Driving Cycle.

3. Fuel economy of the Honda Civic vehicle with the CVCC engine, as measured over the Federal Emissions Test Driving Cycle, was about ten percent lower than that of conventional Civic vehicles, and 16 percent lower than that of equivalent-weight 1973 model year emission certification vehicles. A Vega vehicle with this type of engine showed fuel economy five to ten percent better than that of the standard Vega. A CVCC powered Impala and a Texaco TCCS powered M-151 vehicle showed equal or slightly better fuel economy compared to the respective unmodified vehicles.

4. Both open-chamber and divided-chamber engines have demonstrated a lower sensitivity to fuel octane number than the basic spark ignition engine.

5. A potential problem area for both open-chamber and divided-chamber engines is high production cost. The open-chamber engine also requires cylinder fuel injection. Additional potential problem areas related to open-chamber engines include emission control system durability and the current inability to achieve oxides of nitrogen emission levels that meet original 1976 Federal emission standards without incurring a substantial loss in fuel economy.

SUMMARY

The Clean Air Act Amendments of 1970 were the stimulus for vigorous research efforts on the part of the automotive and related industries to find the means to meet stringent exhaust emission standards for hydrocarbon (HC), carbon monoxide (CO), and nitrogen oxides (NO_x) required for 1975 model year automobiles and beyond. Most of these efforts focused on modifying the combustion process and/or treating exhaust gases of the conventional spark ignition internal combustion engine. Some of the research effort went into investigating the feasibility of low-emission alternative types of engines.

Table S-1 summarizes the applicable statutory emission standards for gasoline-powered passenger cars as currently promulgated.* As can be noted, the Federal standards apply nationwide except for those instances where California has been granted a waiver to adopt more stringent control of one or more emission species in a given time period.

Through model year 1974, these standards have been met by the combination of engine modifications (combustion chamber redesign, spark retardation, etc.) and the incorporation of exhaust gas recirculation systems for control of nitrogen oxides to meet 1973 - 74 standards. In regard to the 1975 California standards, it is generally agreed that the required emission control system is exemplified by the following package of components and engine modifications:

- Oxidation catalytic converters
- Air injection

---

*Exhaust emission goals for engines under investigation in the AAPS Program are set at one-half the 1977 Federal standards. The requirements are made more severe to: (a) account for changes from prototype to mass-production designs, (b) ensure that production engines, with expected statistical variations, can pass the Federal Certification Test, and (c) allow for expected increases in emissions as vehicle mileage reaches the 50,000-mile Federal certification limits.

Table S-1. A Summary of Passenger Car Emission Standards (Gasoline Powered)

Applicable Year and Area		Emission Rate, Grams/Mile															
		Uncontrolled Pre-1966[a]		1966 Cal 1968 Fed	1970 Cal	1970 Fed	1971 Cal	1971 Fed	1972 Cal	1972 Fed	1973 Cal	1973 Fed	1974 Cal	1974 Fed	1975[c] Cal	1975[c] Fed	1976[d] Nation	1977[d] Nation
Test Procedure		FTP	CVS-C	FTP (Concentration Measure)					CVS-C						CVS-CH			
Emission	Hydrocarbons	10	17	3.4	2.2	2.2	2.2	2.2	3.2	3.4	3.2	3.4	3.2	3.4	0.9	1.5	0.41	0.41
	Carbon Monoxide	77	125	35	23	23	23	23	39	39	39	39	39	39	9.0	15	3.4	3.4
	Nitrogen Oxides	4-6	4-6	NR	NR	NR	4	NR	3.2[b]	NR	3.0	3.0	2.0	3.0	2.0	3.1	2.0	0.4
	Evaporative (grams per test)	40		NR	6	NR	6		2		2		2		2		2	2

FTP — 7-mode Federal test procedure

NR — No requirement

CVS-C — CVS cold start mass test

CVS-CH — CVS cold/hot weighting mass test

[a] Estimate only, for comparison purposes

[b] Two hot cycles of FTP test

[c] Original 1975 standards were: HC= 0.41, CO= 3.4, NO_x= 3.1

[d] Original 1976 standards were the same as current 1977 standards

- Exhaust gas recirculation
- Carburetor modifications
- Ignition system modifications.

(There is not complete agreement as to whether the catalytic converter will be required to meet the less stringent Federal 1975 standards which apply to the rest of the nation.) An example of emission control system elements for control of both exhaust and evaporative emissions is given in Figure S-1 for General Motors' projected 1975 design.

Emission control systems to meet the 1976 standards are projected to be similar to the 1975 systems, with improvements in catalytic converters and associated components to meet the lower hydrocarbon and carbon monoxide requirements.

Emission control systems which have been considered by the automobile manufacturers to meet the stringent nitrogen oxide standard prescribed for 1977 include all components of the 1974 system, plus:

- Reduction catalyst(s) installed upstream of the oxidation catalyst(s)
- More sophisticated air injection systems
- Modified carburetion, ignition, and EGR systems.

To date, no manufacturer has reported acceptable durability for reduction catalysts; however, further development testing is in progress.

The ability of the conventional internal combustion engine to meet the original 1976 Federal exhaust emission standards has been the subject of debate for some years. Even with the most elaborate emission control schemes, prospects for meeting these standards are considered marginal by most automakers. Further, the add-on emission control systems are expensive, they affect driveability of the car, their durability for 50,000 miles is a serious problem, and they result in degraded fuel economy.

As an alternative, a number of other engines have been considered for the automobile power plant of the future. These engines, as conceived, may not require add-on devices; in most cases, it is expected that

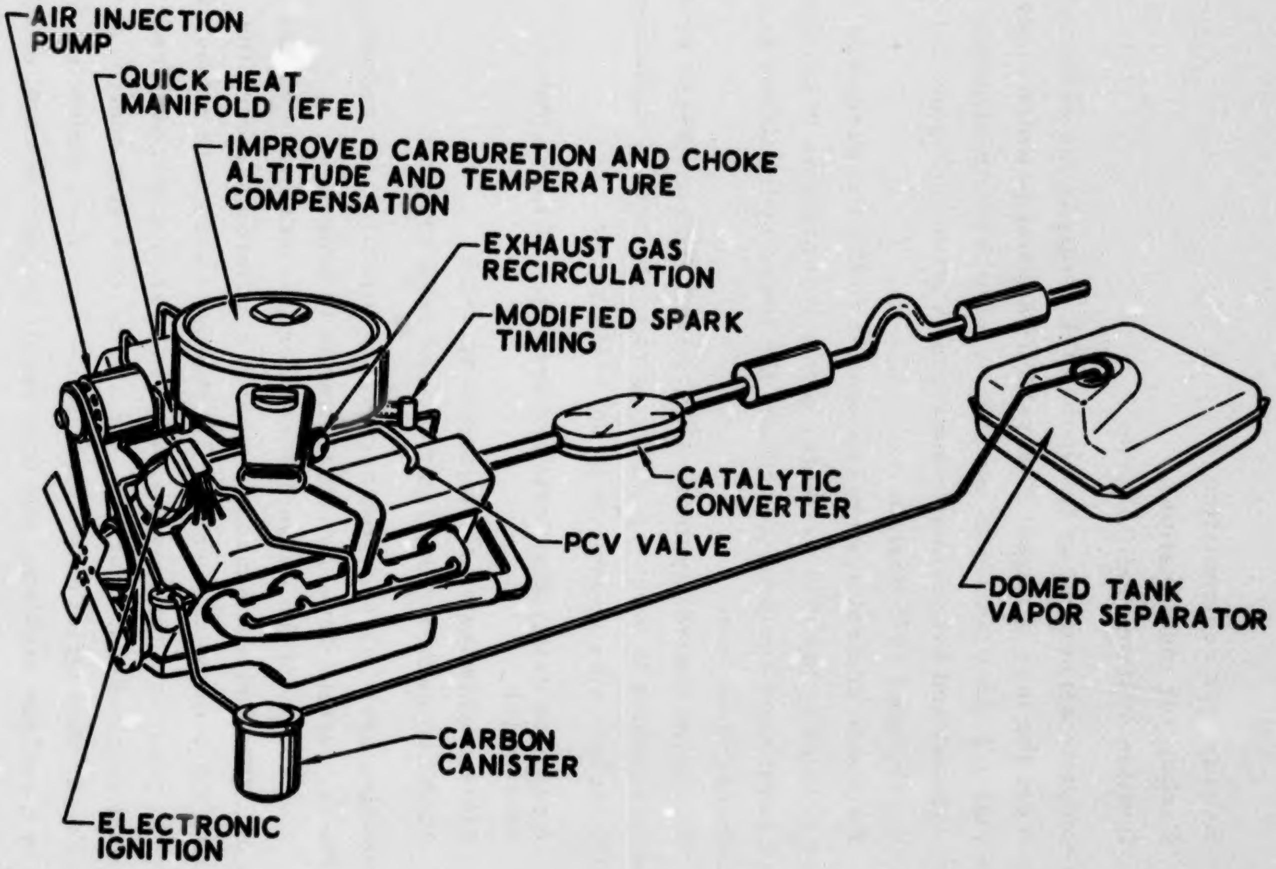

AIR INJECTION PUMP
QUICK HEAT MANIFOLD (EFE)
IMPROVED CARBURETION AND CHOKE ALTITUDE AND TEMPERATURE COMPENSATION
EXHAUST GAS RECIRCULATION
MODIFIED SPARK TIMING
CATALYTIC CONVERTER
PCV VALVE
DOMED TANK VAPOR SEPARATOR
CARBON CANISTER
ELECTRONIC IGNITION

Figure S-1. General Motors Projected 1975 Under-Floor Emission Control System the basic combustion process can be adequately controlled to achieve the expected low emission levels. In this regard, the most promising systems rely on continuous combustion processes as opposed to the intermittent combustion processes found in the spark ignition engine.

Evaluation of low-emission alternative engines to meet the Federal requirements has been a continuing task since 1970, when its program of research and technology development was launched by EPA to evaluate the feasibility of various automotive power plants. The program is called the Alternative Automotive Power Systems (AAPS) Program and is administered by the Alternative Automotive Power Systems Division of EPA. This program was intended to supplement the work of industry, thereby ensuring that all reasonable technical approaches to meeting the standards were being pursued. A few of the engines under study were invented several decades ago and have been used successfully as power generators for a number of years. However, until recent applications of modern-day technology, these concepts were considered too inadequate in weight, size, cost, and efficiency for serious consideration as a mass-produced automobile power plant.

Six types of power systems were initially included in the program: gas turbine, Rankine cycle, heat engine/electric and heat engine/flywheel hybrids, all-electric, and stratified charge engines. In July 1971 it was decided to limit the number of candidate systems under development and to concentrate all available funding and manpower resources on the three most promising systems for the near term: the gas turbine, the Rankine cycle, and stratified charge engines. The theoretical characteristics and anticipated development schedules of these three systems were considered more favorable than the others. Consequently, development work is focusing on these three, with major efforts going to the gas turbine and Rankine cycle engines because of their greater need for new technological advancements. The program is schedule to be completed in 1975 with demonstrations, tests, and evaluations made of Rankine cycle, gas turbine, and stratified charge engines in automobiles.

This volume of the report to the Administrator is intended to summarize the status of engine development work in the AAPS Program and of other similar hardware development programs on alternative engines being funded by private industry as of the end of 1973. The engines under review are the: gas turbine, Rankine cycle, Stirling, diesel, Wankel (and other rotary piston designs), and stratified charge. Of these, gas turbine, Rankine cycle, and Stirling cycle engines operate with continuous combustion processes; whereas, the diesel, Wankel, and stratified charge (and the current spark-ignition) engines operate with intermittent combustion processes. (Continuous combustion processes offer the engine designer more flexibility in tailoring combustion for low emissions.) A discussion of current progress in development of these engines is summarized in the next section.

S.1 PROGRESS IN ENGINE DEVELOPMENT

S.1.1 Gas Turbine Engine

S.1.1.1 General Description

In its simplest form, the gas turbine engine consists of a compressor, a combustion chamber, and a turbine (Figure S-2). Air is taken in by the compressor at atmospheric pressure, compressed to a higher pressure, then delivered to the combustion chamber where fuel is injected and burned at essentially constant pressure. The resulting high-temperature, high-pressure gas is expanded in the turbine and then exhausted to the atmosphere. Part of the turbine-shaft work is used to drive the compressor; the remainder is the output work.

Several modifications of this simple cycle have been employed in the development of gas turbine engines for vehiclar use. Most current engines for on-highway vehicles have a free power turbine (Figure S-3); i.e., two mechanically independent turbine stages are used. One drives the compressor and the other drives the output shaft. This allows a wide range in output-shaft speed, which is not possible with a single-shaft turbine engine.

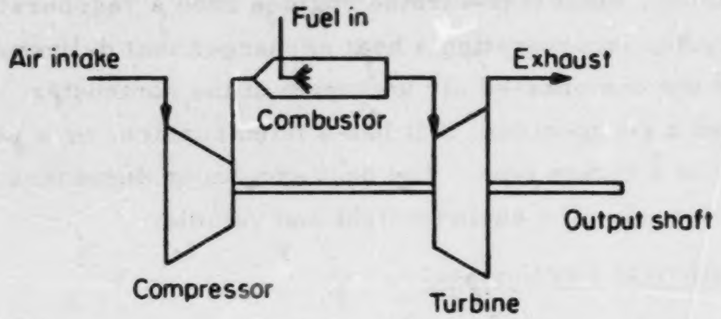

Fuel in
Air intake
Exhaust
Combustor
Output shaft
Compressor
Turbine

Figure S-2. Schematic Diagram of Simple Gas Turbine Engine

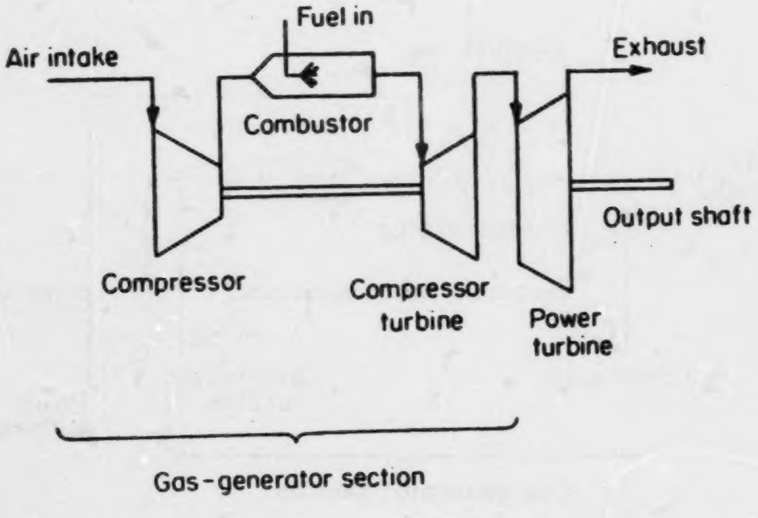

Fuel in
Air intake
Exhaust
Combustor
Output shaft
Compressor
Compressor turbine
Power turbine
Gas-generator section

Figure S-3. Schematic Diagram of Free Turbine Engine

To improve efficiency, most free-turbine engines used a regenerative or recuperative cycle, incorporating a heat exchanger that delivers heat from the exhaust gas to the compressed air upstream of the combustor. The heat exchanger is called a recuperator, if it has a fixed surface, or a regenerator (Figure S-4) if it is a rotary type. The heat exchanger decreases the exhaust temperature but increases the engine weight and volume.

S.1.1.2 Historical Development

The fundamental principles of the gas turbine engine were well known by the end of the eighteenth century; however, turbomachinery operating successfully on the gas turbine thermodynamic cycle, called the Brayton cycle, is a fairly recent development. The first really successful gas turbine power plant was the aircraft turbojet developed from intensive work begun in the 1930s.

Development of gas turbine engines for automotive uses was begun immediately after World War II with Chrysler in this country and

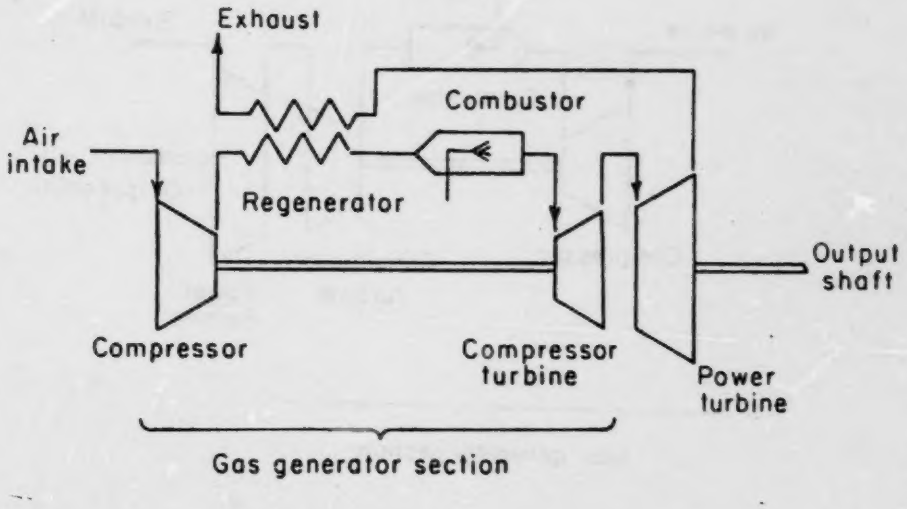

Exhaust
Combustor
Air intake
Regenerator
Output shaft
Compressor
Compressor turbine
Power turbine
Gas generator section

Figure S-4. Schematic Diagram of Regenerative Free Turbine Engine

Rover in Great Britain. The first Rover turbine-power automobile ran in 1949. General Motors began automotive gas turbine work around 1948; Ford about 1950. All of these developers selected the free turbine configuration early and moved quickly to the regenerative configuration. In 1964, Chrysler fabricated and tested a demonstration fleet of 50 gas turbine-powered automobiles. More recently, General Motors and Ford's principal gas turbine engine developments have focused on engines for bus and truck applications instead of for passenger cars. To date, no such units are commercially available; but in some cases prototype engine-vehicle testing is continuing.

In 1971, the AAPS gas turbine program was initiated to attempt to solve the technical problems that had plagued past attempts at developing a production gas turbine for the passenger car. The basic elements of the program are shown in Figure S-5. In this type of program, automotive gas turbine engines are initially used as test beds to establish baseline performance and emission levels. Then they are used to verify component improvements resulting from the technology programs sponsored in the areas of low-emission combustion, manufacturing cost studies, improved part-load fuel economy studies, materials development, and transmission and controls studies. An engine improvement phase is included that allows redesign of the gas turbine according to the technology advancements made. The development team selected to implement the program is shown in Figure S-6.

Estimated high production costs continue to be a major factor inhibiting the implementation of the automotive gas turbine engine. Other problems involve the need for improvements in noise level, durability, acceleration lag, and fuel economy. These problems are being investigated by Chrysler under a $6.5 million EPA contract awarded in December 1972 for the development of an improved version of its sixth-generation engine for powering an intermediate-size car. (Engine design for a compact car will be examined in a recent extension to this program.)

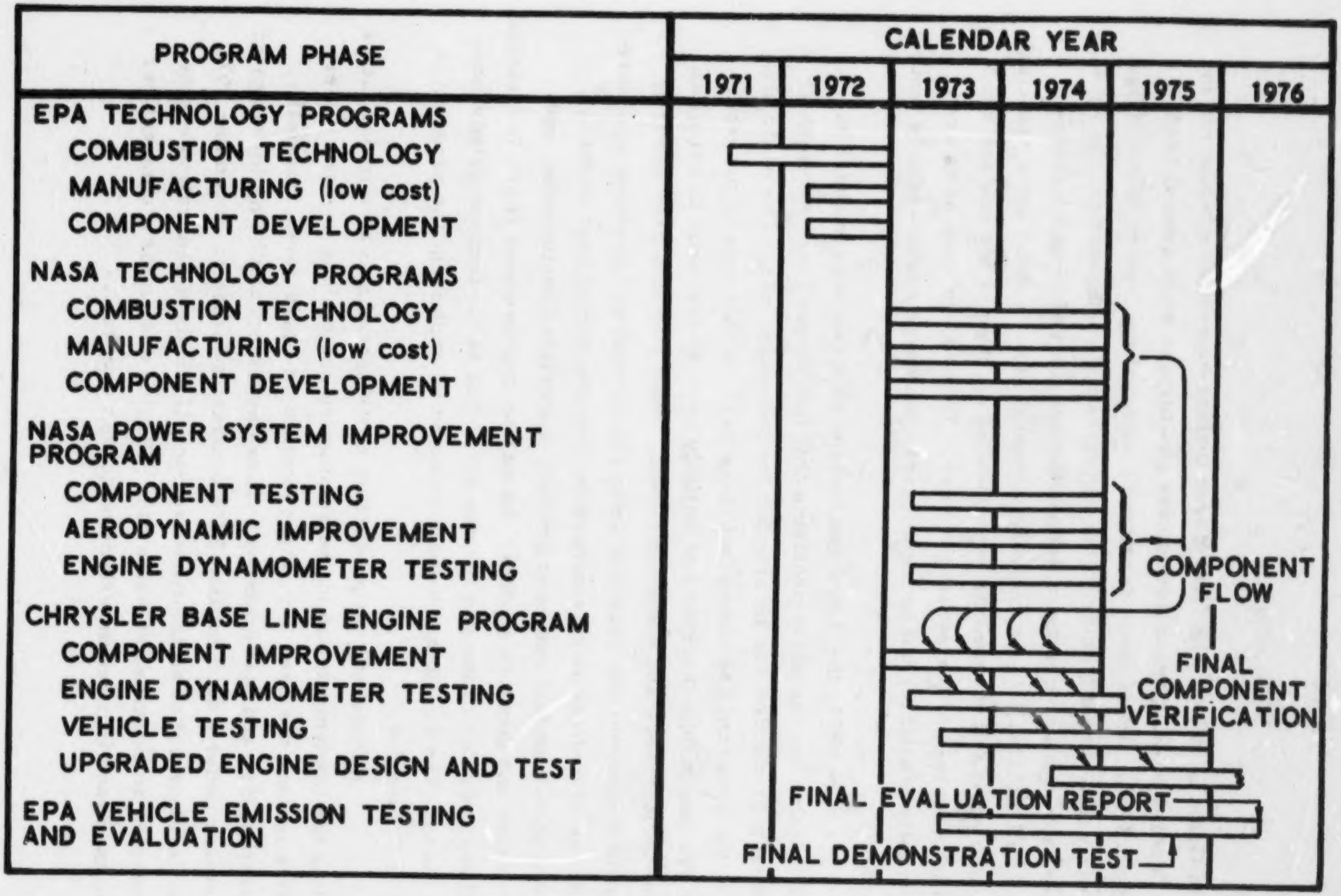


Figure S-5. Gas Turbine Power System AAPS Program

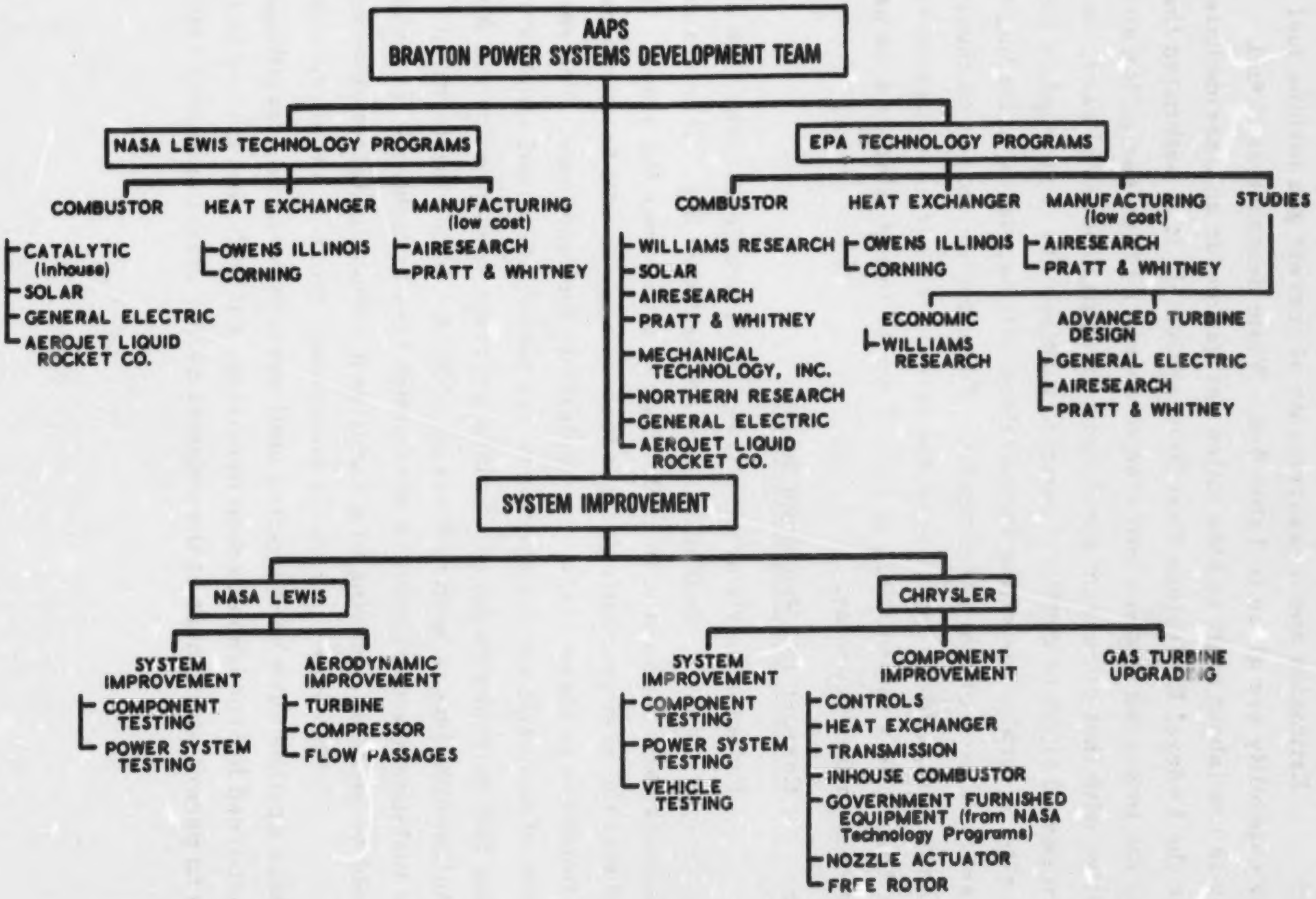

AAPS
BRAYTON POWER SYSTEMS DEVELOPMENT TEAM
NASA LEWIS TECHNOLOGY PROGRAMS
COMBUSTOR
CATALYTIC (inhouse)
SOLAR
GENERAL ELECTRIC
AEROJET LIQUID ROCKET CO.
HEAT EXCHANGER
OWENS ILLINOIS
CORNING
MANUFACTURING (low cost)
AIRESEARCH
PRATT & WHITNEY
EPA TECHNOLOGY PROGRAMS
COMBUSTOR
WILLIAMS RESEARCH
SOLAR
AIRESEARCH
PRATT & WHITNEY
MECHANICAL TECHNOLOGY, INC.
NORTHERN RESEARCH
GENERAL ELECTRIC
AEROJET LIQUID ROCKET CO.
HEAT EXCHANGER
OWENS ILLINOIS
CORNING
MANUFACTURING (low cost)
AIRESEARCH
PRATT & WHITNEY
STUDIES
ECONOMIC
WILLIAMS RESEARCH
ADVANCED TURBINE DESIGN
GENERAL ELECTRIC
AIRESEARCH
PRATT & WHITNEY
SYSTEM IMPROVEMENT
NASA LEWIS
SYSTEM IMPROVEMENT
COMPONENT TESTING
POWER SYSTEM TESTING
AERODYNAMIC IMPROVEMENT
TURBINE
COMPRESSOR
FLOW PASSAGES
CHRYSLER
SYSTEM IMPROVEMENT
COMPONENT TESTING
POWER SYSTEM TESTING
VEHICLE TESTING
COMPONENT IMPROVEMENT
CONTROLS
HEAT EXCHANGER
TRANSMISSION
INHOUSE COMBUSTOR
GOVERNMENT FURNISHED EQUIPMENT (from NASA Technology Programs)
NOZZLE ACTUATOR
FREE ROTOR
GAS TURBINE UPGRADING

Figure S-6. EPA/AAPS Brayton Power Systems Development Team

Estimates and measurements of current gas turbine fuel economy capability are given in Table S-2. While demonstrated fuel economy is low (about eight to nine miles per gallon for an intermediate-size car over the Federal Emissions Test Driving Cycle), it is estimated that fuel economy for improved engines will rise to about 12 miles per gallon and be competitive with that of current spark ignition engine-powered cars. In the far-term period (1985 to 2000), if ceramic turbines are developed, a substantial rise in turbine operating temperature will be possible, leading to a significant reduction of fuel consumption. For example, based on theoretical estimates, a temperature increase at the turbine inlet from a current value of about 1800°F to a future value of 2500°F would result in about a 20 percent improvement in fuel economy.

S.1.1.2 Current and Projected Status

Principal early automotive-turbine problems were high cost, noise, poor fuel economy, durability, and acceleration lag. Considerable improvements have been made in each of these areas over the years. It is expected that the next generation of gas turbine engines will be somewhat further improved in these areas, particularly in fuel economy. The recent awareness of domestic energy shortages has placed additional emphasis on improving fuel economy in the gas turbine program sponsored by the Alternative Automotive Power System Division of EPA. In the far term, if ceramic turbines are developed, a substantial rise in turbine inlet temperature would be possible, leading to a significant reduction in fuel consumption.

With regard to exhaust emissions, those of automotive-type gas turbine engines have been typified until recently as characteristically low in unburned hydrocarbon/carbon monoxide, but with nitrogen oxide levels too high to permit attainment of the original 1976 Federal emission standard

Table S-2. Gas Turbine Fuel Economy Data Summary

Engine	Rated Power Level at 59°F	Am-bient Temp., °F	Fuel Economy, mi/gal							Vehicle or Test Weight, lb
			FETDC	20 mph	30 mph	40 mph	50 mph	60 mph	70 mph	
United Aircraft Research Laboratories[a]										
RGSS-6 (Regenerative Single Shaft)	150	59	12.16	15.06	18.02	18.65	17.96	16.59	14.70	4000
RCSS-8 (Recuperative Single Shaft)	150	59	11.20	13.73	16.59	17.33	16.73	15.42	13.79	3950
SS-10 (Simple Cycle)	150	59	7.51	8.86	11.21	12.51	12.63	12.22	11.47	3700
AiResearch[a]										
Free-turbine Regenerated	175	59	12.5	16.1	18.2	18.2	17.2	15.5	13.7	4000
Chrysler										
Free-turbine Regenerated (sixth-generation engine)	150	--	8.5[b]	--	--	18	--	--	--	Inter-mediate size car
Williams Research										
WR26 Engine Free-turbine Regenerated	80	--	5.1, 8.1[b]	--	--	--	--	--	--	AMC Hornet (compact)

[a] Estimates based on computer simulation.

[b] Based on tests over Federal Emissions Test Driving Cycle (FETDC).

mi/gal = miles per gallon
mph = miles per hour

of 0.4 grams per mile. For example, Chrysler's sixth-generation engine installed in an intermediate-size vehicle has nitrogen oxide emissions of about 2.2 grams per mile over the Federal Emissions Test Driving Cycle. However, within the AAPS Program, Solar, a Division of International Harvester, has recently developed a new gas turbine combustion system that, based on steady-state test data, is calculated to be capable of meeting the original 1976 Federal emission standards and is close to meeting the AAPS Program goals (one-half the level of the original standards). Additional development is required to perfect the combustor control system and to validate meeting program goals with transient operation of the engine over the Federal Emissions Test Driving Cycle.

Reflecting similar recent advancements a General Motors-sponsored program resulted in an experimental 225-horsepower gas turbine engine installed in a car meeting the original 1976 Federal emission standards. Complete details concerning engine design and operation are not available. The available gas turbine data are summarized in Table S-3.

Estimates of the manufacturing costs of gas turbine engines in mass production differ widely. A recent cost analysis conducted under the AAPS Program indicated somewhat higher initial turbine engine cost, as compared to a piston engine, but a potential for a slightly lower cost with further development. While encouraging, it is recognized that actual costs are typically higher than projected costs by the time the hardware is developed, tested, and produced. The study nevertheless provided valuable insight into processes that must be developed and components whose designs and materials must be simplified and improved in order to have a low-cost engine. Several cost-reduction programs are under way for processing current metal components. The successful development of ceramics for stationary hot parts and, in the longer term, for rotating parts would lower the production cost of the gas turbine significantly and conserve the country's scarce resources of alloying materials.

Table S-3. Gas Turbine Exhaust Emission Data Over the Federal Emissions Test Driving Cycle

Engine/Manufacturer	Emissions (gm/mi)			Remarks
	CO	HC	NO_x	
a. Conventional Combustors				
United Aircraft Research Laboratories RGSS-6	0.53	0.15	2.72	Calculated values over simulated Federal Driving Cycle, based on emission data from GM engine GT-309.
United Aircraft Research Laboratories SSS-10	1.86	0.31	1.03	Calculated values over simulated Federal Driving Cycle, based on emission data from GM engine T-56.
Chrysler sixth-generation engine	3.99	-0.26	2.21	1975 Federal Test Procedure emissions corrected (background level subtracted from measured values).
Williams Research/ WR26/AMC Hornet	7.43 6.92	0.62 0.72	2.8 2.5	Cold start Hot start } 1975 Federal Test Procedure
Williams Research/ 131Q/Volkswagen	4.5	0.34	1.81	1972 Federal Test Procedure.
b. Advanced Combustors				
Solar	3.34	0.11	0.34	Calculated values for simulated Federal Driving Cycle, based on advanced combustor data.
General Motors 225 HP Regenerative Engine	2.4	0.015	0.315	Experimental test bed engine, chassis dynamometer test with 5,000-lb car, 1975 Federal Test Procedure.
AiResearch	1.6	0.67	0.44	Calculated values for simulated Federal Driving Cycle, for recuperated engine with 10 percent bypass.
United Aircraft of Canada	3.64	0.49	0.52	Calculated values over simulated Federal Driving Cycle based on advanced combustor data.
c. Clean Air Act Requirements	3.4	0.41	0.4	Original 1976 Federal standards

S.1.2 Rankine Cycle Engine

S.1.2.1 General Description

The Rankine cycle engine is an external combustion engine in which high-pressure steam or some other working fluid is expanded in a reciprocating (piston-type) or turbine-type device to produce work. Components of a typical Rankine cycle engine are shown in schematic form (Figure S-7). The feed pump draws liquid at a low pressure from the reservoir and forces it under pressure into the vapor generator, where it is converted to superheated vapor by heat from the combustion gases. The hot, high-pressure vapor then flows into the expander where it gives up energy as it expands to a lower pressure and temperature. Exhaust vapor from the expander enters the condenser, where it is converted back to a liquid by rejecting heat to the surroundings. The resulting low-pressure liquid is then returned to the reservoir or sump to complete the cycle. The major components shown (feed pump, vapor generator, expander, condenser) are required for any Rankine cycle system. Specific systems will show variations in such features as type of drive for the auxiliaries, additional heat transfer paths from one part of the cycle to another, and the type and location of liquid reservoir and boost pump.

S.1.2.2 Historical Development

Rankine cycle steam cars held a prominent place in the early days of the automobile. Steam cars were sold commercially in the late 1800s and early 1900s with familiar names such as Stanley Steamer, Locomobile, White, and Doble. These early steam cars had a simple pressurized boiler containing a large quantity of water, which required about a 30-minute cold startup and posed a continual safety problem. They were noncondensing, dumping exhaust steam overboard. Consequently, frequent stops were required for makeup water. In spite of these disadvantages, some of these cars, such as the Stanley Steamer, were quiet running and performed quite satisfactorily. Various technical advances were achieved through the years,

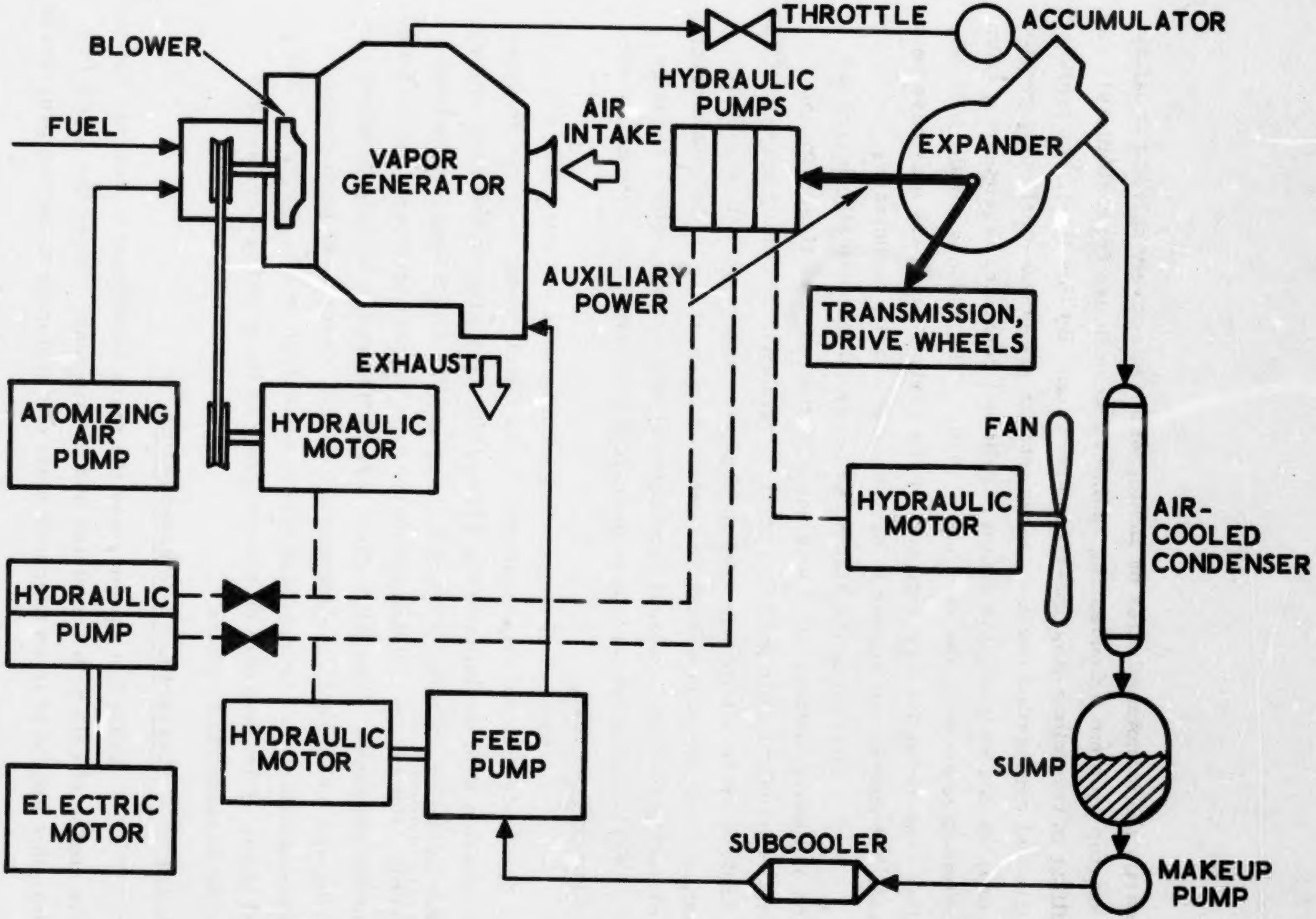

BLOWER
FUEL
VAPOR GENERATOR
AIR INTAKE
THROTTLE
ACCUMULATOR
HYDRAULIC PUMPS
EXPANDER
AUXILIARY POWER
TRANSMISSION, DRIVE WHEELS
EXHAUST
ATOMIZING AIR PUMP
HYDRAULIC MOTOR
FAN
HYDRAULIC MOTOR
AIR-COOLED CONDENSER
HYDRAULIC PUMP
ELECTRIC MOTOR
HYDRAULIC MOTOR
FEED PUMP
SUMP
SUBCOOLER
MAKEUP PUMP

Figure S-7. Rankine Cycle System Schematic primarily the monotube boiler by Doble, which effectively solved the safety and startup problem. Condensing radiators were in use from about 1915 on, resulting in true close-Rankine-cycle operation. By this time, however, the internal combustion engine entered into its era of low-cost mass production and, in this respect, the steam car did not compete, disappearing from the automotive scene in the very early 1930s. The reasons for the lack of similar cost-reduction development for the early steam cars may have been varied, but they do not appear to have been primarily technical.

Interest in the steam car occurred sporadically during the fifties and early sixties; but it was not until the latter sixties, with the impact of impending air pollution control legislation becoming apparent, that the Rankine cycle automobile system began to receive serious scientific attention. The impetus behind this renewed interest lay in the clean-burning characteristics of the external combustor of this system, which requires neither post-engine exhaust cleanup devices nor compromises with an efficient burning process.

A number of engineering efforts were undertaken at about this time by widely diverse organizations. This development received impetus and cohesion with the initiation in 1970 of the EPA Alternative Automotive Power System Program. Figure S-8 depicts the AAPS Rankine program schedule, and Figure S-9 delineates the development team selected to implement the program. This EPA effort was structured to include problem solving and component development activities. Component performance improvements were incorporated into preprototype engine systems. The final phase of development on this program is the prototype system and vehicle demonstration phase.

S.1.2.3 Current and Projected Status

Table S-4 summarizes the more significant worldwide Rankine cycle engine efforts both completed and in progress. Under the AAPS Program, this engine is in an advanced state of development, having progressed

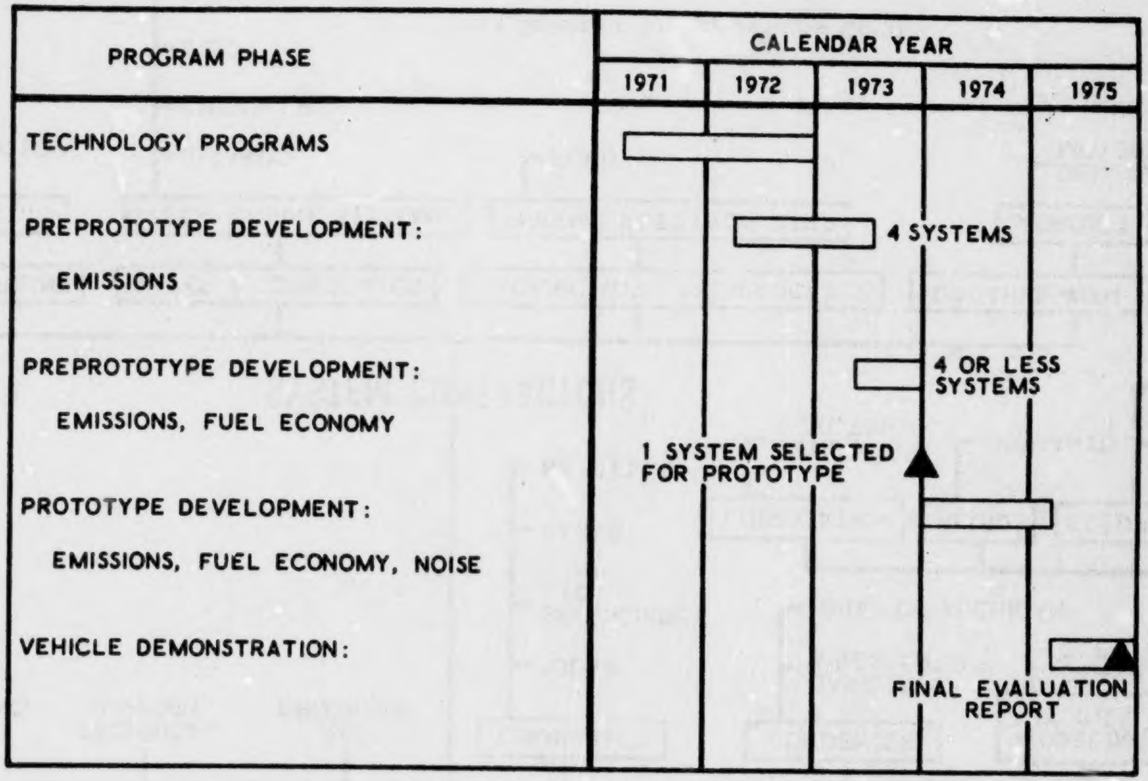
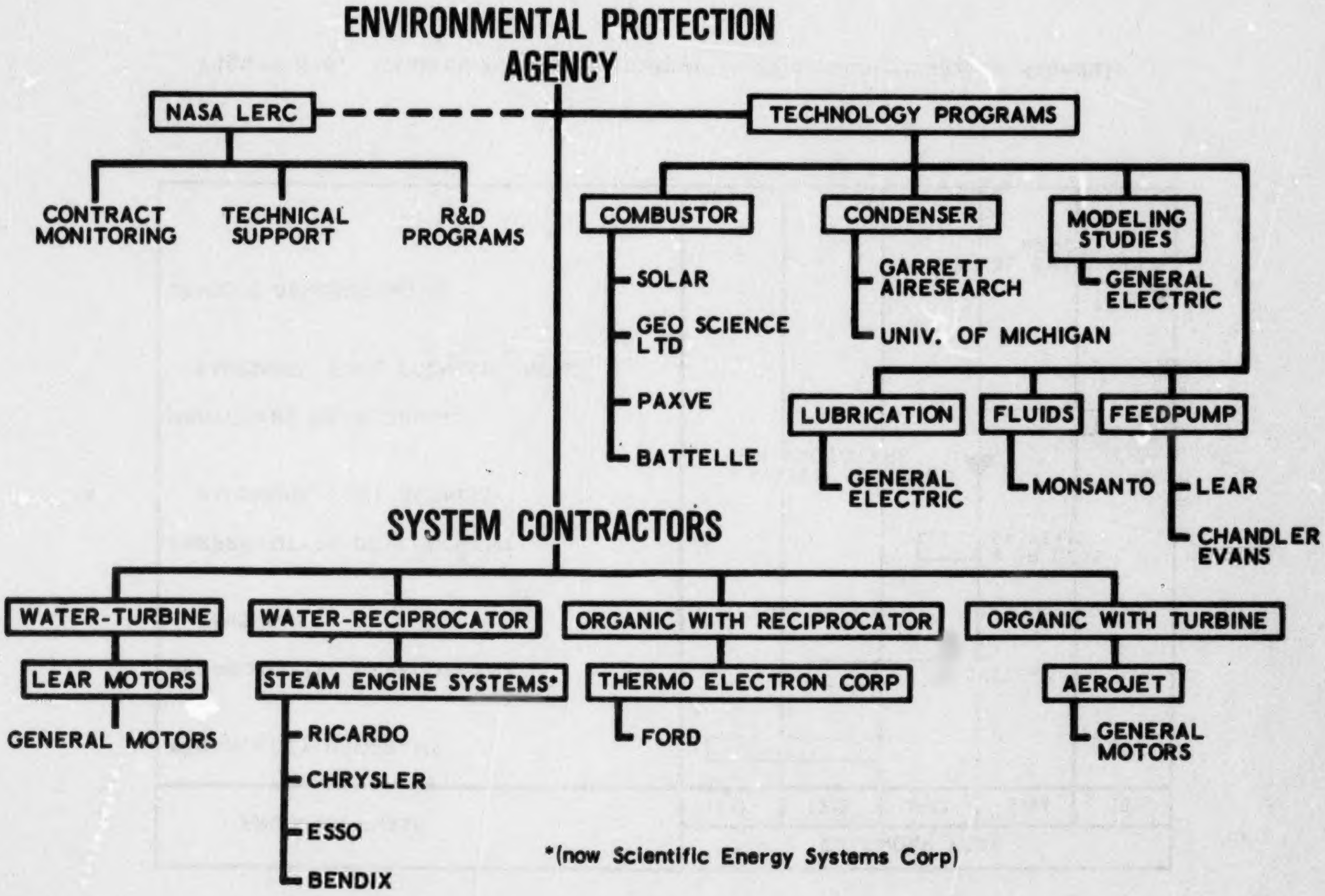


Figure S-8. Rankine System Prototype Development Program Schedule


ENVIRONMENTAL PROTECTION AGENCY
NASA LERC
CONTRACT MONITORING
TECHNICAL SUPPORT
R&D PROGRAMS
TECHNOLOGY PROGRAMS
COMBUSTOR
SOLAR
GEO SCIENCE LTD
PAXVE
BATTELLE
CONDENSER
GARRETT AIRESEARCH
UNIV. OF MICHIGAN
MODELING STUDIES
GENERAL ELECTRIC
LUBRICATION
GENERAL ELECTRIC
FLUIDS
MONSANTO
FEEDPUMP
LEAR
CHANDLER EVANS
SYSTEM CONTRACTORS
WATER-TURBINE
LEAR MOTORS
GENERAL MOTORS
WATER-RECIPROCATOR
STEAM ENGINE SYSTEMS*
RICARDO
CHRYSLER
ESSO
BENDIX
ORGANIC WITH RECIPROCATOR
THERMO ELECTRON CORP
FORD
ORGANIC WITH TURBINE
AEROJET
GENERAL MOTORS

*(now Scientific Energy Systems Corp)

Figure S-9. EPA/AAPS Rankine System Development Team

Table S-4. Summary of Rankine Cycle Engine/Vehicle Development Programs

Developer/Contractor	Vehicle	Rated (or Estimated) Horsepower	Type of Expander and Rated Speed in rpm	Working Fluid	Status of Program
EPA AAPS Program					Preprototype engine development program completed in December 1973. One contractor (SES) selected to proceed with prototype engine development, followed by installation in vehicle.
Aerojet Liquid Rocket Co.	6-Passenger car[a]	150, Expander gross	Turbine; 32,000	AEF-78 (Proprietary)	
Lear Motors Corp.	5-Passenger car[a]	123, Expander gross	Turbine; 65,000	Water	
Scientific Energy Systems	6-Passenger car[a]	158, Expander gross 138, Transmission in	Piston; 2,500	Water	
Thermo Electron Corp.	6-Passenger car[a]	146, Expander gross	Piston; 1,800	Fluorinol - 85	
DOT/California Steam Bus Program					Program completed September 1972. Consisted of demonstration use in public service and vehicle emissions testing.
Brobeck & Associates	51-Passenger bus	240, Gross bhp 200, Net bhp	Piston; 2,100	Water	
Lear Motors Corp.	51-Passenger bus	240, Gross bhp 180, Net bhp	Turbine; 65,000	Water	
Steam Power Systems Co.	51-Passenger bus	275, Gross bhp 224, Net bhp	Piston; 1,850	Water	
DOT/Dallas Program					Program completed. Consisted of service in transit system and vehicle emissions testing.
Sundstrand Corp.	25-Passenger bus	120, Expander gross 90, Gear box out	Turbine; 35,000	CP-25; a single chemical compound comprised of methyl benzene, toluene, and toluall.	
California Clean Car Project					Testing and vehicle installation in progress. Target date for vehicle delivery is May 15, 1974.
Aerojet Liquid Rocket Co.	4-Passenger car[a]	60, Expander gross	Turbine; 60,000	Water	
Steam Power Systems Co.	4-Passenger car[a]	65, Expander gross	Piston; 2,400	Water	
Other Domestic Programs					
Carter Enterprises, Inc.	4-Passenger car (Volkswagen)	70	Piston; 5,000	Water	Vehicle first driven March 1972. Performance measurements being made.
Kinetics, Inc.	4-Passenger car	—	Gerotor type	Refrigerant 113	No published emissions or fuel economy data.
Williams Engine Co.	4-Passenger car	—	Piston	Water	
Foreign Programs					
Pritchard Steam Power Proprietary, Ltd.	6-Passenger car	—	Piston	Water	
SAAB-SCANIA	Passenger car[a]	160	Piston	Water	Component development. Recently initiated.
Politecnico Milano	Bus[a]	100	Turbine	Organic	

[a] Simulated or installation design target. bhp - brake horsepower DOT - U S. Department of Transportation

to the point where several complete engine systems have been tested on engine dynamometers. (Other programs listed in Table S-4 have produced demonstration vehicles that have shown system feasibility.) Some of these engine systems are water-base working fluid designs (steam engine) that present the problem of fluid freezing at low temperatures. Others rely on an organic-type working fluid as one means of circumventing this problem. A major breakthrough in design of the engine condenser has resulted in much smaller units, which has led to improved engine installation in current automobile engine compartments.

Preliminary analyses of steady-state data from the AAPS Program show that exhaust emissions from the Rankine cycle engine may meet the original 1976 Federal standards and that fuel economy can be competitive with current emission-controlled spark ignition engines. Table S-5 summarizes available emission estimates for four classes of Rankine cycle engines. Figure S-10 compares both current experimental and projected Rankine cycle fuel economy characteristics with values obtained with two prototype conventionally powered 1975 model year automobiles.

The task of repeated demonstrations of low emissions in transient operation (including cold start) with a fully automatic control system has been initiated under the AAPS Program. Scientific Energy Systems Corporation (formerly Steam Engine Systems) has been selected from several competing contractors to provide EPA with a vehicle-installed steam engine system for testing. This engine design utilizes a reciprocating piston expander for transmitting power through a transmission to the rear wheels (in contrast to other designs based on a turbine expander).

Future AAPS Program efforts are expected to concentrate on achieving improvements in fuel economy and investigating low-cost production techniques.

Table S-5. Preliminary Rankine Cycle Emission Results Over the Federal Emissions Test Driving Cycle

Engine Class	Emissions,[a] gm/mi		
	HC	CO	NO_x
Water Reciprocating	0.09	0.50	0.20
Water Turbine	0.21	0.70	0.38
Organic Reciprocating	0.02	0.19	0.25
Organic Turbine	0.16	1.01	0.13

[a] These values are based upon steady-state measurements and are analytically converted to the 1972 Federal Test Procedure.

gm/mi = grams per mile

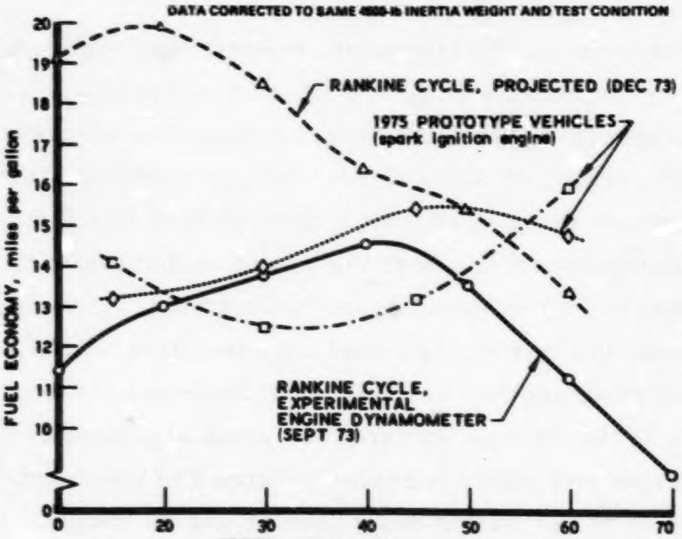


Figure S-10. Comparative Fuel Economy for Rankine Cycle Engine

S.1.3 Stirling Engines

S.1.3.1 General Description

The Stirling engine is an external-combustion, closed-cycle, piston-type power plant that uses a gaseous internal working fluid, usually hydrogen or helium (see Figures S-11 and S-12). Cyclical heating and cooling varies the pressure of the fluid within a closed volume, the pressure variations being transmitted to a piston, thereby developing output power. Heat transfer processes are accomplished at high efficiency by alternately and regeneratively moving the working fluid between heater and cooler sections. Thus, the engine contains at a minimum one expansion volume, one compression volume, one regenerator, one heater, and one cooler.

The ideal efficiency of the Stirling engine is 100 percent of the Carnot efficiency obtained between peak and minimum fluid temperatures. Various losses reduce actual efficiencies to about one-half of the ideal value. Nevertheless, the Stirling engine is one of the most efficient prime movers known.

S.1.3.2 Historical Development

Not a recent development, the Stirling-cycle engine was patented in 1816 by Robert Stirling, a Scottish clergyman. In 1938, N. V. Philips Laboratories at Eindhoven, Holland, undertook the development of a modern version of this engine. Its research resulted in the development of the rhombic drive displacer engine in 1953 and the roll-sock seal in 1960. This configuration had both the power piston and displacer piston in a single cylinder. The rhombic drive made straight line rod movement possible and opened the way to high working pressures and high power. Displacer-type Stirling engines are operating in buses and boats in Europe.

In 1970, Philips initiated the most significant advancement yet by achieving a major reduction in engine volume and complexity, coupled with a major increase in power output per pound of engine weight. This occurred through the design of and successful operation of a double-acting Stirling engine with swashplate drive mechanism. This engine consists of

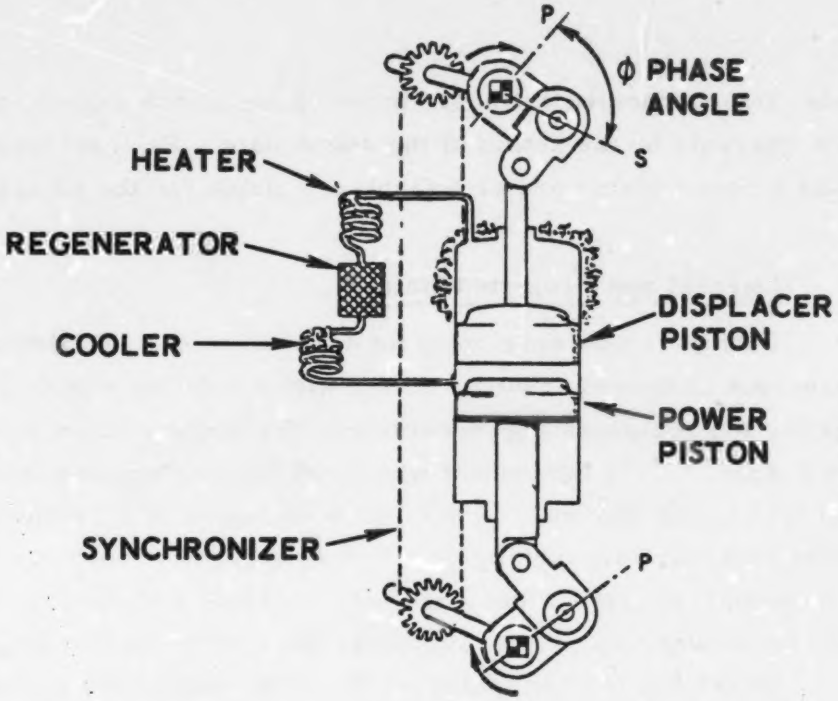

P
ϕ PHASE ANGLE
S
HEATER
REGENERATOR
COOLER
DISPLACER PISTON
POWER PISTON
SYNCHRONIZER
P

Figure S-11. Schematic, Basic Stirling Displacer Engine

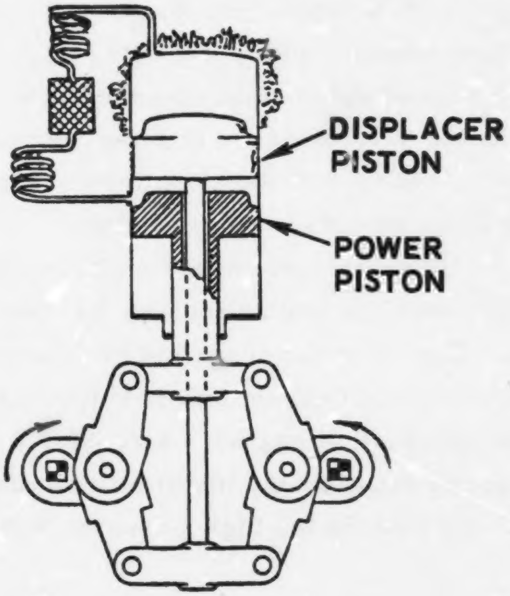

DISPLACER PISTON
POWER PISTON

Figure S-12. Schematic, Rhombic Drive Displacer Engine four separate, interconnected cylinders in which the piston motion is phased at 90-degree intervals by the action of the swashplate. Each piston serves alternately as a power piston and as a displacer piston for the adjacent cylinder.

S.1.3.3 Current and Projected Status

Philips is now developing an automotive engine in which the four cylinders are equally spaced around a circle with a rotating wobble (or swash) plate converting the reciprocating movement of the pistons into a rotating motion (Figure S-13). A 1971 agreement with Ford Motor Company is currently scheduled to lead to the demonstration of such an engine in a Torino car in 1975. United Stirling in Sweden is working on a V-4 configuration with conventional connecting rods and crankshaft that Ford will install in a Pinto car. M.A.N/MWM in Germany is also working on improvements and cost reduction designs.

Incentives for continuing work on this engine are excellent fuel economy, multifuel capability, very low noise and vibration, size and weight comparable to current engines, and emissions potentially low enough to meet the original 1976 federal standards. Table S-6 compares projected Stirling engine fuel economy characteristics with those estimated for a spark ignition engine-powered automobile equipped with 1976-type emission control systems. Table S-7 compares Stirling engine exhaust emissions (simulated) with the original 1976 Federal emission standards. The projected emission levels fall well below the standards.

Primary problem areas with the Stirling engine requiring further development work for resolution are the radiator size and the expense of heater tubes. For efficient operation of this engine, the radiator in certain designs can be about two and one-half times as large as that for a comparable internal combustion engine. A reduction in radiator size will be necessary to permit satisfactory installation in current automobile engine compartments. To contain the high-pressure hydrogen working fluid, the

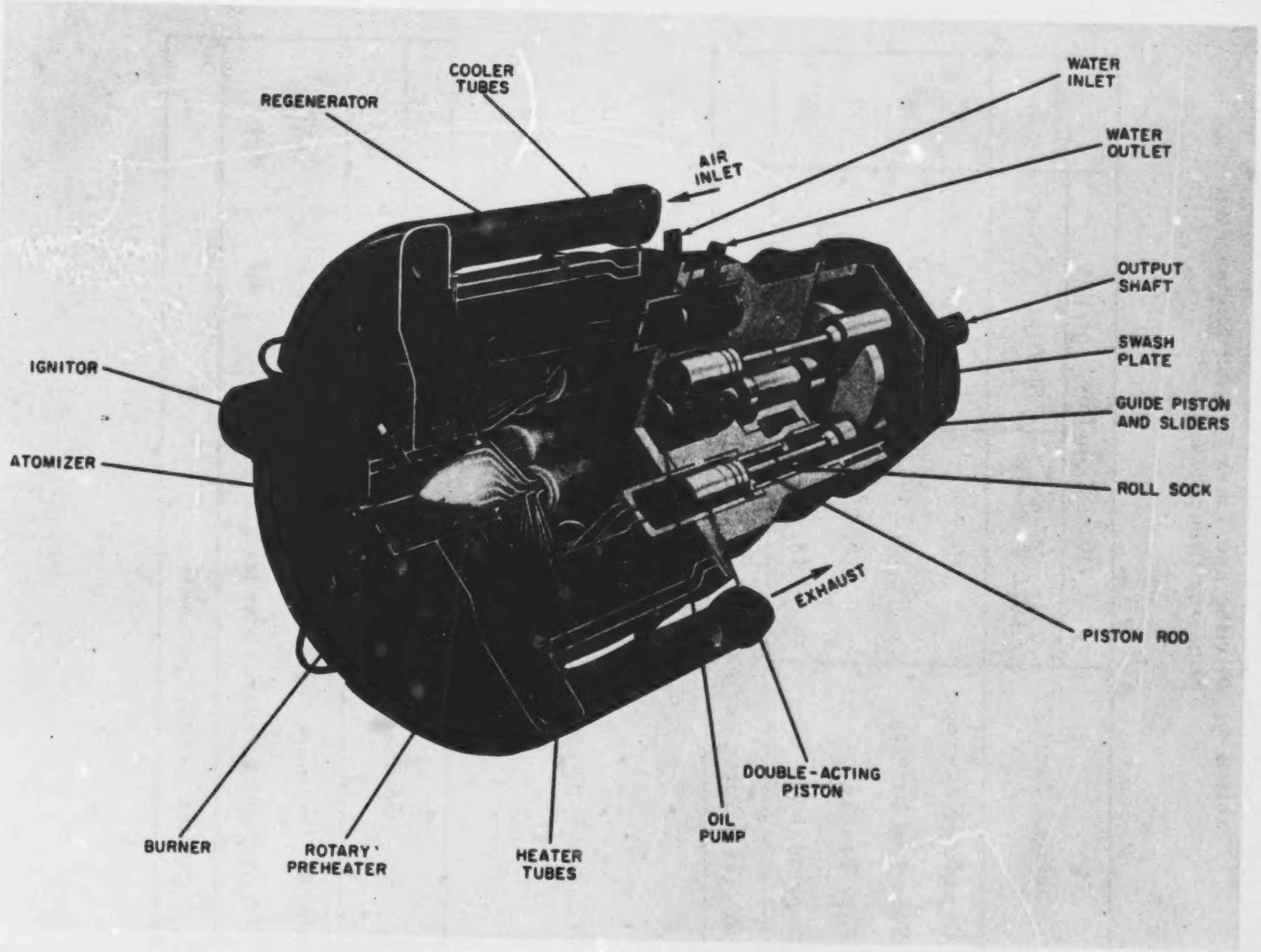

REGENERATOR
COOLER TUBES
AIR INLET
WATER INLET
WATER OUTLET
OUTPUT SHAFT
SWASH PLATE
GUIDE PISTON AND SLIDERS
ROLL SOCK
IGNITOR
ATOMIZER
EXHAUST
PISTON ROD
BURNER
ROTARY PREHEATER
HEATER TUBES
OIL PUMP
DOUBLE-ACTING PISTON

Figure S-13. Philips Stirling Double-Acting Engine with Swash-Plate Drive

Table S-6. Stirling and Spark Ignition Engine Fuel Economy Comparisons

Driving Cycle	Fuel Economy, mi/gal		Gain, %
	1976 Ford V8	Stirling-Torino (Projected)	
Consumer Average	10.7	14.7	37
Suburban Driving	13.3	17.9	35
City Traffic	8.2	11.4	39
Superhighway	13.0	16.6	28
mi/gal = miles per gallon			

Table S-7. Constant Volume Sampling Test Simulation Emissions

Comparison	Emissions, gm/mi		
	HC	CO	NO_x
Philips Engine	0.10	0.31	0.175
Original 1976 Federal Standard	0.41	3.40	0.4
gm/mi = grams per mile			

array of heater tubes is now made of nickel-chrome alloys. These tubes are expensive to fabricate and, therefore, production processes must be improved or new designs must evolve to aid in reducing engine manufacturing costs.

Other problems with the Stirling engine include inadequate piston seal life, hydrogen diffusion through the cylinder walls and seals, and the need for low-cost power control capable of providing smooth power transitions. Engineering solutions are available, but they require further development and demonstration.

S.1.4 Diesel Engine

S.1.4.1 General Description

The diesel or compression ignition engine is a reciprocating engine which is, in many respects, very similar to the Otto cycle spark ignition engine. In the diesel, air alone is compressed in the cylinder, and the fuel is then injected into the heated air towards the end of the compression stroke. The temperature developed during compression is sufficiently high to ignite the fuel immediately after injection. Thus, spark plugs, distributor, and carburetor are eliminated in the diesel; but a high-pressure fuel-injection system is required. Since compression-ignition demands high-pressure ratios, the current diesel engines are heavier, more bulky, and costlier than equivalent spark ignition engines.

The principal advantages of diesels relative to spark ignition engines include their ruggedness and their higher thermal efficiency, which results in lower fuel consumption, particularly at part-load operating conditions.

Many different engine configurations have been built around the basic thermodynamic diesel cycle, and a number of these are currently being marketed by many companies throughout the world. The various designs can be grouped into two categories -- open chamber and divided chamber -- and each of these can then be subdivided into different engine classes, depending upon specific features related to the design of the combustion chamber and the induction system. Open-chamber designs are favored by most manufacturers of heavy-duty diesels because of the better fuel economy obtained with this engine type. Conversely, divided-chamber designs (prechambers and swirl chambers) are preferred by the manufacturers of light-duty engines because of their higher speed capability.

S.1.4.2 Historical Development

The diesel engine was invented by Rudolf Diesel in 1892 and was originally intended to burn coal dust. In his early design, Diesel proposed to coordinate the rate of fuel injection with the movement of the pistons in such a manner that the heat of combustion would be liberated at constant temperature, thus approaching the thermal efficiency of the Carnot cycle. However, this approach proved to be unsuccessful and the cycle was then modified to achieve near-constant pressure combustion. While this ideal process is still being utilized in large, low-speed diesels, the high-speed diesels used in light-duty applications employ a process that lies between the ideal cycle and the Otto cycle.

The high compression ratio required to achieve fuel ignition, and the resulting bulk, precluded the application of diesels in early road vehicles. The first diesel-powered truck was built in 1907 by Saurer in Switzerland, in cooperation with Diesel; and in 1911 the first fully loaded diesel truck crossed the United States coast to coast.

Daimler-Benz of Germany started series production of its Mercedes Benz 260D diesel-powered automobile in 1936. Since resumption of production in 1949, Daimler-Benz has sold more than one million diesel automobiles. A number of other companies have since entered the light-duty diesel engine field, including Peugeot of France, Perkins of England, Daihatsu of Japan, and Opel of Germany.

S.1.4.3 Current and Projected Status

Because of their favorable fuel economy characteristics, diesel engines are widely used in Europe and Japan as power plants for taxicabs, passenger cars, and other light-duty vehicle applications. However, the power output of these engines is considerably lower than that required for the heavier American vehicles, and their duty cycles are generally quite different.

The Mercedes Benz 220D diesel-powered automobile has been marketed in the United States since 1956. Between 1956 and 1972, Daimler-Benz sales of diesel cars in the United States amounted to about 53,000 units, with more than 6,000 sold in 1972 alone. In 1971, Daimler-Benz produced about 284,000 automobiles and 99,000 of these were diesel-powered 220Ds.

Peugeot of France has introduced its diesel-powered 504-passenger car in the United States in the spring of 1974. Since production began in 1959, Peugeot has sold more than 480,000 diesel-powered, light-duty vehicles. In 1972, Peugeot's diesel sales amounted to about 80,000 units out of a total production of 671,000 vehicles.

Based on test data taken by EPA, research institutions, and engineering/manufacturing firms, light-duty vehicles powered by diesel engines have emissions below the original 1975 Federal emission standards (Table S-8), but there are no indications as yet that this engine can adequately meet the original 1976 Federal oxides of nitrogen standard. Additionally, diesel odor and particulates (smoke) have long been recognized as very undesirable exhaust emission products. Progress toward determination of the cause of the odor has been very slow, though its control has been achieved in some engines through fuel injection system refinements.

The fuel economy of diesel-powered vehicles over the Federal Emissions Test Driving Cycle is between 50 percent and 70 percent better than that achieved by the average of 1973 model year emission certification vehicles tested at the same inertia weight. Table S-9 lists some specific comparative data for diesel- and gasoline-powered vehicles. Currently, however, because installed diesel engine power is relatively low, vehicle acceleration is significantly lower than that found in cars powered with spark ignition engines. On an equivalent performance basis, the fuel economy advantage of a higher-powered diesel vehicle would be expected to be smaller than that for current lower-powered diesel vehicles.

Table S-8. Average Light-Duty Diesel Vehicle Emissions (1975 Federal Test Procedure)

Test Vehicle	Test Laboratory	Emissions, gm/mi		
		HC	CO	NO_x
Mercedes Benz 220D	EPA	0.34	1.42	1.43
Mercedes Benz 220D (modified injection)	EPA	0.28	1.08	1.48
Mercedes Benz 220D	Daimler-Benz	0.15	2.3	1.45
Opel Rekord 2100D	EPA	0.40	1.16	1.34
Peugeot 504 Diesel	EPA	3.11	3.42	1.07
Peugeot 504 Diesel	Peugeot	2.50	2.20	1.30
Ford/Nissan Diesel	EPA	1.70	3.81	1.71

gm/mi = grams per mile

Table S-9. Light-Duty Diesel and Gasoline Vehicle Fuel Economy Comparisons (1975 Federal Test Procedure)

Diesel Powered		Gasoline Powered	
Vehicle	Fuel Economy, mi/gal	Vehicle	Fuel Economy, mi/gal
Mercedes Benz 220D (standard injection)	23.6	1973 Mercedes 220	13.2
Mercedes Benz 220D (modified injection)	24.6	1973 Mercedes 220	13.2
Opel Rekord 2100D	23.8	---	---
Peugeot 504	25.2	1973 Peugeot 504	17.0
Ford/Nissan Pickup	21.4	---	---
---	---	1973 Average 3000-pound car	15.6
---	---	1973 Average 3500-pound car	13.9
---	---	1973 Average 4500-pound car	10.1

mi/gal = miles per gallon

Power output per pound of engine weight is considerably lower than that presently offered in spark ignition engine-powered domestic automobiles. Methods for increasing the power output per pound of engine weight include (a) means to increase power by turbocharging or supercharging and (b) means to reduce engine weight and bulk by a reduction in engine compression ratio and a shortened design life.

S.1.5 Wankel and Other Rotary Piston Engines

S.1.5.1 General Description

In its simplest form, the Wankel engine consists of a single triangular-shaped rotor which revolves eccentrically in a chamber with a double-lobed cross section (Figure S-14). The motion of the rotor is such that the apexes of the rotor continuously brush the walls of the chamber, thereby forming three compartments which rotate around the chamber, changing size and shape as they move by virtue of the eccentricity of rotation. Thermodynamically, the engine is based on the Otto cycle in its "four-stroke" form. On one side of the chamber casing is a spark plug. Each compartment passes successively through all phases of the Otto cycle once in each rotor revolution. Since there are three compartments, three power strokes are produced with each rotor's revolution.

S.1.5.2 Historical Development

Many efforts have been expended in the past century to develop internal combustion engines that will produce a torque at a reasonably low speed without need for a crankshaft and its attendant complications. Within the last decade or so, these efforts have produced a wide variety of prototype rotary piston engines, largely experimental in nature.

Numerous rotary engine design concepts have been proposed, but the one receiving the greatest attention for automotive application is the Wankel Rotary Piston Engine invented by Felix Wankel in 1953. The prototype was run successfully at the NSU Works in Germany on 1 February 1957. Since then, experimental engines of various sizes have been built and

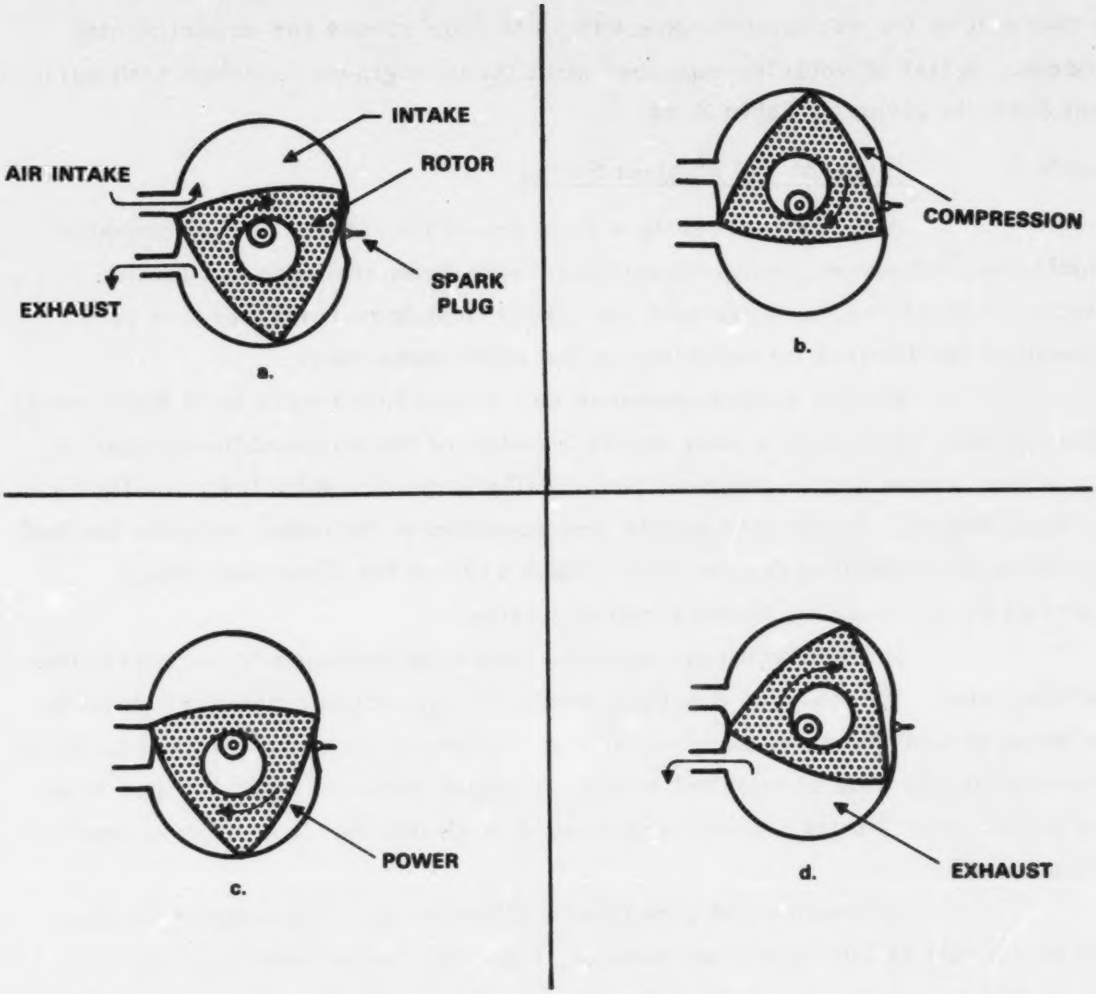

INTAKE
AIR INTAKE
ROTOR
SPARK PLUG
EXHAUST
a.
COMPRESSION
b.
POWER
c.
d.
EXHAUST

Figure S-14. Wankel Rotary Engine

tested for a wide variety of applications covering automobiles, aircraft, boats, snowmobiles, lawn mowers, and auxiliary power units. Current automotive rotary piston engines are water-cooled and are designed with one or two rotors for production cars and up to four rotors for experimental models. A list of vehicles equipped with these engines, together with pertinent data, is given in Table S-10.

S.1.5.3 Current and Project Status

Although currently a very small fraction of the automobile population, rotary engine-powered cars have been rising in production levels. The Chevrolet Vega with the General Motors rotary engine is scheduled for limited introduction in the 1975 model year.

Rotary engine-powered car production levels have been rising, although they represent a very small fraction of the automobile population. Currently, these engines appear principally in Mazda vehicles manufactured by Toyo Kogyo. Production levels are expected to increase with the limited introduction scheduled for the 1975 model year of the Chevrolet Vega powered by the General Motors rotary engine.

A manufacturing problem exists in attempts to increase production rates. Because of complex machining operations required on both the housing and rotor, production of this engine is currently limited to about 25 per hour per line (compared to piston engine rates of over 100 per hour per line). This limitation appears to have a significant impact on manufacturing costs.

Technological problems still exist for some engine designs. These consist of high gas seal leakage, high thermal stresses, poor low-speed fuel economy, and high exhaust emissions without aftertreatment. With aftertreatment (a thermal reactor), the Mazda rotary engine has demonstrated the ability to meet the original 1975 Federal emission standards (Table S-11).

Table S-10. Vehicles Equipped with Rotary Piston Engines

Vehicle	Introduction	Discontinued	Engine	Horse-power	Rotors	Compression Ratio
NSU Wankel Spider	1964	1967	NSU	64	1	8.6:1
NSU Ro-80	1967	—	NSU	136	2	9.0:1
Mazda Cosmo 110-S	1967	1969	Toyo Kogyo	110	2	9.4:1
Mazda R-100	1969	—	Toyo Kogyo	110	2	9.4:1
Mazda RX-2	1970	—	Toyo Kogyo	120	2	9.4:1
Citroen M-35	1970	Limit 500 cars	NSU	55	1	9.0:1
Mercedes Benz C-III MK I	1969	Experimental	Mercedes Benz	330	3	9.3:1
Mercedes Benz C-III MK II	1970	Experimental	Mercedes Benz	400	4	9.3:1
Mustang RC 2-60	1965	Experimental	Curtiss-Wright	185	2	8.5:1

Table S-11. Best Emissions Results, Thermal Reactor and Wankel Engine, 2750-Pound Compact Car

Manufacturer	System[a]	Mileage	No. of Tests	Emissions,[b] gm/mi			Fuel[c] Penalty, %
				HC	CO	NO_x	
Toyo Kogyo	Reactor	---	1973 Certification	2.4	20.0	0.9	0
Toyo Kogyo	Reactor	4,000	Many	0.32	3.1	0.83	6.5
		50,000	3	0.36	2.6	0.87	--
	Reactor + EGR	Low	8	0.35	2.2	0.49	12.0
General Motors	Reactor	Low	72	0.60	5.0	0.60	--
	(Best effort)	Low	1	0.43	2.8	0.44	--

[a] All these systems are carbureted fuel rich (air-fuel ratio of about 12).

[b] 1975 Constant volume sampling, cold/hot test procedure (grams per mile).

[c] Relative to 1973 production rotary engine car.

S.1.6 Stratified Charge Engines

S.1.6.1 General Description

In principle, the stratified charge engine represents a modification of the conventional spark ignition engine. The principal objective of the modification is the achievement of heterogeneous combustion in which a rich fuel-air mixture is generated around the spark plug and a lean mixture in the remaining zones of the combustion chamber. The resulting two-stage combustion process permits operation of the engine at very lean overall air-fuel ratios that are conducive to low emissions, good fuel economy, and reduced sensitivity of the engine to fuel octane and cetane numbers.

The stratified charge engines can be divided into two distinct classes: open-chamber engines (Figure S-15) and divided-chamber engines (Figure S-16). In the open-chamber configurations, exemplified by the Texaco TCCS and Ford PROCO engines, a single combustion chamber is used similar to that of conventional spark ignition engines. Upon ignition of a swirling, rich mixture surrounding the spark plug, the burning charge expands into the power regions of the combustion chamber where the combustion process is then completed in an oxygen-rich environment.

The divided-chamber stratified charge, or prechamber, engines, exemplified by Honda's CVCC engine concept, employ two interconnected combustion chambers per cylinder. During the compression stroke of the piston, a fuel-rich mixture is inducted into the generally smaller prechamber while the main chamber is charged with a lean mixture or even with pure air. Upon ignition in the prechamber, hot gases expand into the main chamber where combustion is then carried to completion. The principal advantage of prechamber engines over conventional engines is their ability to operate with very lean overall fuel-air mixtures resulting in concurrently low hydrocarbon, carbon monoxide, and oxides of nitrogen emissions.

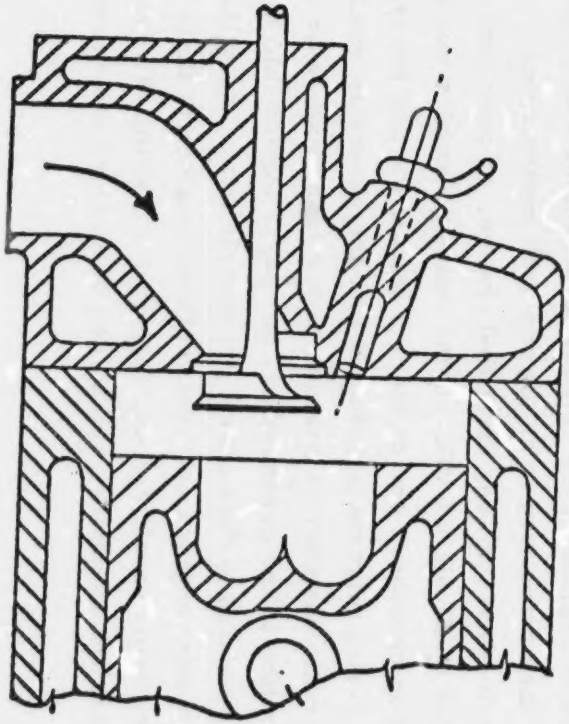


Figure S-15. Texaco Cup Combustion, Open-Chamber Stratified Charge Engine

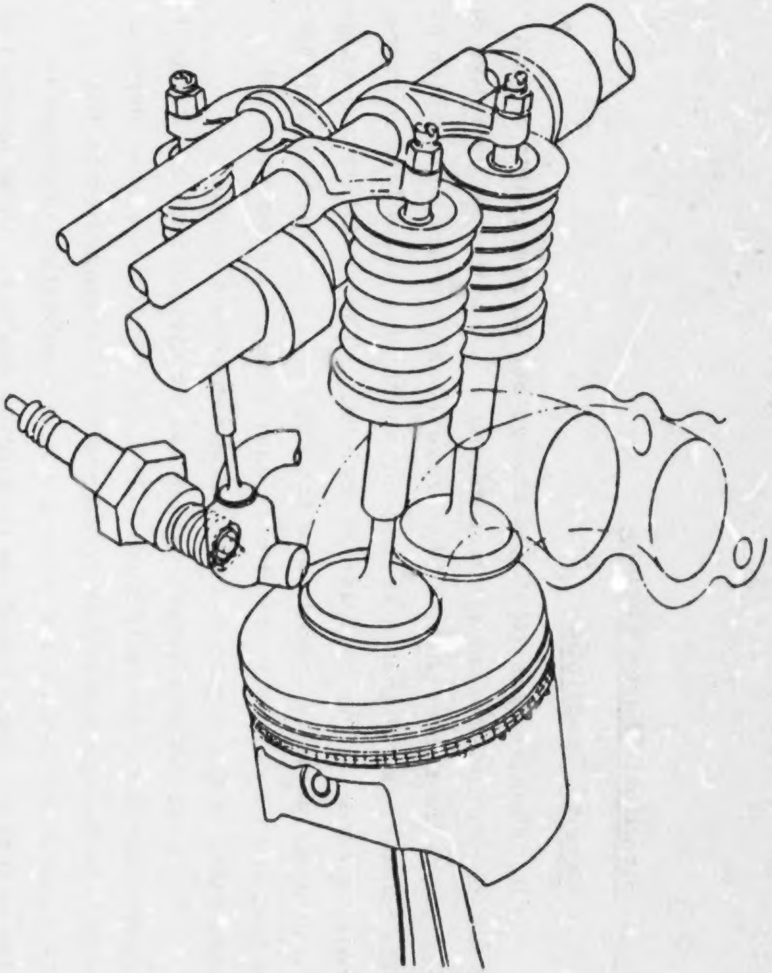


Figure S-16. Honda CVCC, Divided-Chamber Stratified Charge Engine

S. 1. 6. 2 Historical Development

The concept of stratified charge operation of internal combustion engines and its potential was first evaluated in the early 1920's by Ricardo in England. In the United States, work on open-chamber configurations commenced in the 1940's with Texaco's invention of a "knockless" engine. In 1965, the United States Army Tank Automotive Command (TACOM) initiated funding to Texaco and Ford for exploratory research of both open-chamber and divided-chamber stratified charge engine concepts directed toward developing a military engine with multifuel capability and improved fuel economy relative to conventional spark ignition engines.

In 1970, the relatively low exhaust emissions of the available stratified charge engines and the apparent potential for improvement led to additional funding by EPA through the Army. The TACOM program was a joint Army/EPA-funded effort. Figure S-17 shows the AAPS Program schedule for stratified charge engine development. Analytical and test evaluations of the manufacturing and operational (performance, reliability, maintainability) aspects of the current designs were emphasized to aid in preparing improved design specifications for the next generation of engines.

The concept of a divided combustion chamber can be traced back to the first "oil engines" in the pre-Diesel era. These early prechamber engines operated on heavy oils or naphtha, which was injected into a spherical prechamber and vaporized in contact with the red-hot thermally insulated prechamber walls. Later, attempts were made to apply the prechamber concept to the spark ignition gasoline engine, as documented by numerous patents. However, to date a practical widespread application of the prechamber concept has been realized only in high-speed diesel engines, principally due to the pioneering efforts of Ricardo. A number of organizations and individuals are currently working to develop prechamber

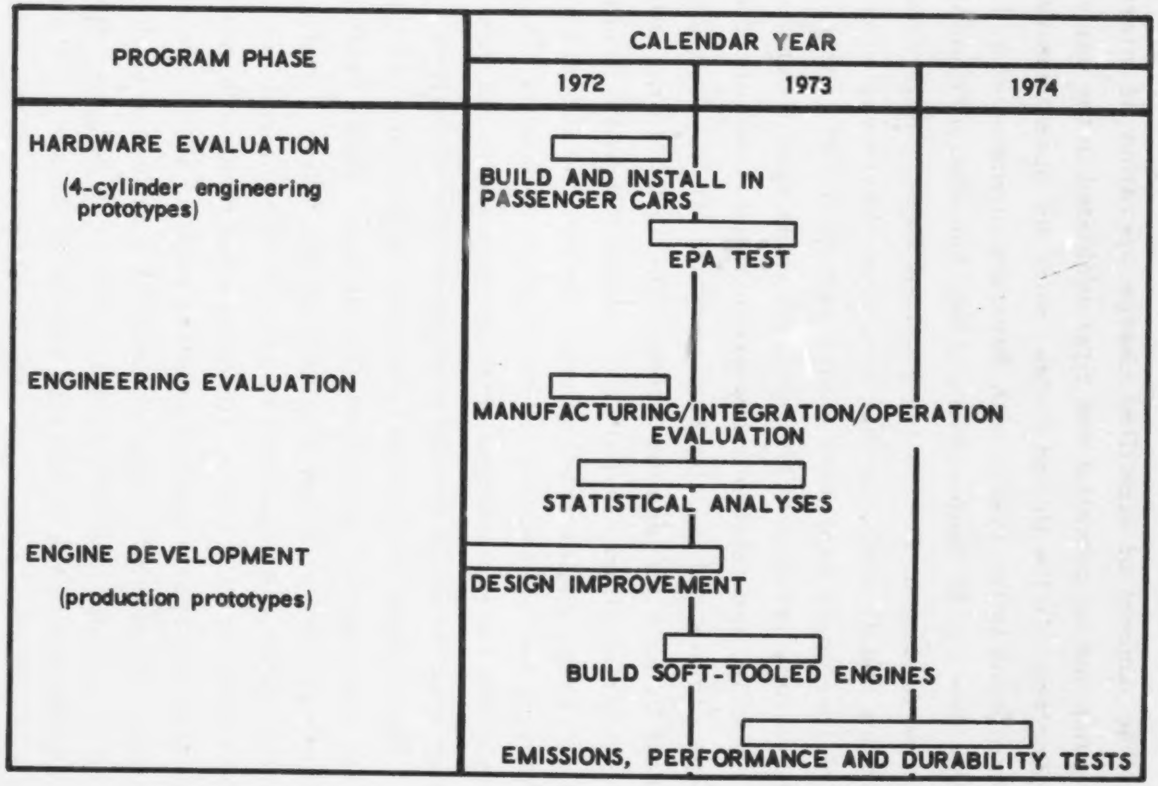


Figure S-17. Stratified Charge System Army/EPA Development Program engines. In particular, during the past several years, the Honda Motor Company of Japan has conducted extensive design, development, and test work on its CVCC concept.

S.1.6.3 Current and Projected Status

Stratified charge engines developed to date, notably by Ford, Texaco, and Honda, have achieved an advanced state of development.

Without incorporation of additional emission control systems, the hydrocarbon and nitrogen oxide emissions from TACOM M-151 vehicles adjusted for best fuel economy and equipped with experimental Ford and Texaco open-chamber stratified charge engines are comparable to those from vehicles powered by conventional spark ignition engines, while carbon monoxide emissions and fuel economy are substantially improved. Conversely, with emission control (consisting of exhaust gas recirculation, oxidation catalysts, and intake air throttling at low power levels), the vehicles meet the original 1976 Federal emission standards at low mileage and show equal or slightly better fuel economy than the average 1973 spark ignition engine-powered certification vehicles tested at equivalent inertia weights. But these M-151 vehicles were unable to negotiate the high acceleration modes of the Federal Emission Test Driving Cycle. Table S-12 presents comparative data for the M-151 vehicle equipped with the Ford PROCO stratified charge engine; similar comparative data are shown in Table S-13 for the Texaco TCCS engine.

Honda has a divided-chamber version (designated CVCC) of the stratified charge engine in production. Without incorporation of additional emission control systems, its CVCC engine-powered Civic vehicle met the original 1976 Federal emission standards for HC and CO, while NO_x was about twice the standard (Table S-14). Vega and Impala vehicles achieved similar emission levels at low mileage when powered by General Motors engines modified by Honda to the CVCC configuration.

Fuel economy of the Honda Civic vehicle with the CVCC engine, as measured over the Federal Emissions Test Driving Cycle, was

Table S-12. Emissions and Fuel Economy, M-151 PROCO Vehicles

Vehicle	Emissions,(3) gm/mi			Fuel Economy,(1)(4) mi/gal	Number of Tests (averaged)	Test Facility
	HC	CO	NO_x			
PROCO, no catalyst	2.60	13.45	0.32	21.7	1	Ford
PROCO, with catalyst	0.35	1.01	0.35	21.3(2)	4	Ford
PROCO, with catalyst	0.37	0.93	0.33	Not measured	14	EPA
1972 CVS Test Procedure						
PROCO, best fuel economy, no catalyst	4.96	7.75	3.85	23.8	2	Ford
PROCO, no catalyst	3.10	13.75	0.33	21.2	1	Ford
PROCO, with catalyst	0.54	1.18	0.37	19.6	4	Ford

CVS = constant value sampling.

(1) Computed from the mass emission data (miles per gallon).

(2) 20.4 miles per gallon, measured with a burette.

(3) Baseline (carbureted) vehicle emissions are 4.55 gm/mi HC, 41.6 gm/mi CO, 4.4 gm/mi NO_x.

(4) Baseline (carbureted) vehicle fuel economy 17.2 mi/gal.

about ten percent lower than that of conventional Civic vehicles, and 16 percent lower than that of equivalent-weight 1973 model year emission certification vehicles. A Vega vehicle with this type of engine showed fuel economy five to ten percent better than that of the standard Vega. A CVCC powered Impala and a Texaco TCCS-powered M-151 vehicle showed equal or slightly better fuel economy compared to the respective unmodified vehicles.

Both open-chamber and divided-chamber stratified charge engines have demonstrated a lower sensitivity to fuel octane number than the basic spark ignition engine. Because of the late fuel injection combined with immediate spark ignition, the Texaco TCCS engine can operate also on a wide range of diesel fuels.

Table S-13. Effect of Emission Control System Modifications on M-151 TCCS Vehicle Emissions and Fuel Economy (1975 Federal Test Procedure)

Engine Configuration Tested at Texaco	Turbo-charged	Emissions,[a] gm/mi			Fuel Economy, mi/gal
		HC	CO	NO_x	
Max. economy setting, no catalyst	Yes	3.13	7.00	1.46	24.3
Max. economy setting, no catalyst	No	4.24	7.28	1.43	--
8 deg. combustion retard only, no catalyst	Yes	3.24	6.43	1.29	22.4
8 deg. combustion retard, low EGR rate	Yes	3.60	6.69	0.84	20.5
8 deg. combustion retard, medium EGR rate, two platinum catalysts	Yes	0.33	1.05	0.61	19.7
13 deg. combustion retard, high EGR rate two platinum catalysts	Yes	0.35	1.41	0.35	16.2
5 deg. combustion retard, no EGR, two platinum catalysts, alternate vehicle	Yes	0.30	1.07	1.40	20.9
Baseline configuration[b]	No	0.36	0.61	0.31	16.2
Baseline configuration,[b] with reduced EGR	No	0.48	0.57	0.45	17.6
Baseline configuration[b,c]	No	0.37	0.24	0.31	15.8
Baseline configuration,[b,c] with reduced EGR	No	0.50	0.14	0.70	21.9

[a] Fuel: gasoline, 91 research octane number, plus 2% oil.

[b] EGR plus two platinum catalysts.

[c] Tested at EPA.

EGR = exhaust gas recirculation
mi/gal = miles per gallon (by weight)
gm/mi = grams per mile

Table S-14. Emissions and Fuel Economy, Honda Civic CVCC Vehicles

Vehicle	Emissions,[a] gm/mi			Fuel Economy, mi/gal	
	HC	CO	NO_x	1975 FTP	1972 FTP
Low-mileage car (#3652. Five tests.)					
Average	0.18	2.12	0.89	22.1	21.0
Maximum[b]	0.21	2.28	1.05	22.4	21.5
Minimum[b]	0.15	1.96	0.75	21.9	20.6
Low-mileage car (#3606. One test.)	0.23	2.00	1.03	20.7	19.5
50,000-Mile car (#2034. Four tests.)					
Average	0.24	1.75	0.65	21.3	19.8
Maximum[b]	0.26	1.85	0.73	22.2	20.0
Minimum[b]	0.19	1.70	0.57	20.8	19.5
Original 1975 Federal standards	0.41	3.40	3.1		
Original 1976 Federal standards	0.41	3.40	0.40		

[a] 1975 Federal Test Procedure (FTP) (grams per mile).

[b] The values of the different columns do not necessarily correspond to the same test.

mi/gal = miles per gallon

The production cost of open-chamber stratified charge engines is expected to be higher than that for conventional engines because of the required fuel injection system, and for the divided-chamber version, a small cost increase over conventional engines is indicated because of the more complex cylinder head configurations.

The open-chamber stratified charge engines fabricated to date are experimental models not suitable for mass production. A number of potential problem areas required further evaluation, including emission control system durability, and the current inability to achieve oxides of nitrogen emission levels that meet original 1976 Federal emission standards without incurring a substantial loss in fuel economy. Ford feels that limited production of PROCO engines might be feasible by 1977. Conversely, the prechamber configuration developed by Honda is currently in limited mass production.

S.2 COMPARATIVE REVIEW OF ALTERNATIVE HEAT ENGINES

Each of the engines being considered herein as an alternative to the conventional spark ignition engine for automobile propulsion has its own unique characteristics. If adequate performance data were available, the engines could be compared to provide a relative ranking of these characteristics. In general, such data are limited in scope or not available for this purpose, because most of these engines are in an early state of development where more effort has been devoted to establishing proof-of-principle for a particular design concept than to establishing all of the prototype baseline performance characteristics. Therefore, in most cases, comparisons are at best subjective.

Of overriding importance, however, is that each alternative engine offers the potential of meeting the original 1976 Federal emission standards, and most show reasonable progress toward these standards as shown by the representative data for selected engines in Table S-15. It should be stressed that because the engines are at different stages of technological development, it is not meaningful to attempt to compare and rank them with regard to the potential for achieving the lowest possible emissions, Table S-15. Selected Exhaust Emission Data, Alternative Heat Engines

Engine	Exhaust Emissions,[a] gm/mi HC	CO	NO_x	Remarks
Gas Turbine				
Chrysler engine	-0.26	3.99	2.21	Background level subtracted from measured values
Solar advanced combustor	0.11	0.9	0.17	Calculated, based on combustor tests
General Motors 225-horsepower	0.02	2.4	0.32	Chassis dynamometer tests, experimental engine
Rankine Cycle				
Water Reciprocating	0.09	0.5	0.2	Calculated, based on component tests (1972 FTP)
Stirling	0.10	0.31	0.18	Calculated, based on component tests
Diesel				
Mercedes Benz 220D	0.34	1.42	1.43	EGR and engine modifications required to reduce NO_x emissions
Wankel				
Toyo Kogyo	0.35	2.2	0.49	Low-mileage tests with thermal reactor and EGR
Stratified Charge				
Honda CVCC	0.24	1.75	0.65	Average values for 50,000-mile test car (Civic)
Ford M-151 PROCO	0.37	0.93	0.33	Based on EPA tests with catalyst and EGR

[a] 1975 Federal Test Procedure (FTP) except where noted (grams per mile).

EGR = Exhaust gas recirculation.

whether for a single exhaust emission specie (e.g., hydrocarbons) or all three species.

Besides emissions, other topics of concern in selecting an engine to supplant the spark ignition engine are: adaptability to mass production manufacture, purchase and maintenance costs (including durability factors), fuel consumption, fuel compatibility, weight and size impact on vehicle packaging requirements, flexibility and responsiveness to driver commands, noise, etc. Current development efforts are directed toward engine size and weight reductions that will permit engine installation in conventional automobile engine compartments. Additional effort is also applied to ensuring engine flexibility and responsiveness and to reducing noise levels.

Mass production manufacturing and the resulting purchase cost are factors that have been reviewed in screening the various alternative engines, but no firm quantitative comparisons are available except in a few specific cases. According to a production implementation study conducted for the U.S. Department of Transportation, dual-shaft, regenerative gas turbines are estimated to cost about twice as much as current spark ignition engines (estimated retail costs ranged from about $550 to $650 for a spark ignition engine). Under low production rates, the cost of current diesel-powered cars ranges from approximately $200 to $700 more than current spark ignition engine-powered cars as cited in congressional hearings conducted in 1973 with respect to suspension of the 1975 Federal emission standards. To provide a proper basis of reference, however, there must be an accounting of the added cost to the consumer of exhaust emission control equipment for the spark ignition engine designed to meet 1976 Federal emission standards. Based on a study conducted for EPA on the effects of lead additives in gasoline on emission control systems for 1975-1976 motor vehicles, these costs are expected to range from approximately $200 to $400.

With respect to fuel consumption, the data are sufficient to afford an illustrative comparison among various engines. Measured fuel economy results for spark ignition engines, contrasted with measurements and estimates of fuel economy for alternative engines, are presented in Figure S-18. As can be seen, some of these alternative engines appear capable of delivering superior fuel economy, particularly when judged against the degraded fuel economy of exhaust-emission-controlled 1973 model year cars. Even on this basis, an absolute comparison is difficult because the fuel economies of the internal combustion engines change with model year.

The compatibility of each engine with various types of fuels can be a strong factor in its selection, particularly for the period beyond 1985 when advanced engines combined with synthetic fuels may be considered practical and significant contributors to national personal transportation.

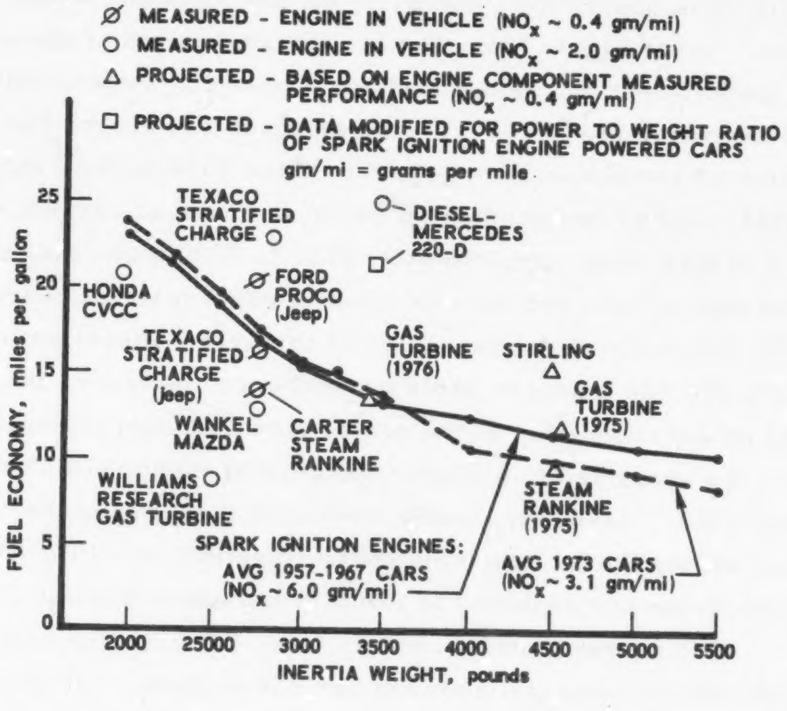


Figure S-18. Fuel Economy Over the Federal Emissions Test Driving Cycle

Those engines utilizing continuous combustion processes (e.g., gas turbine, Rankine cycle, Stirling) offer the widest latitude in the use of alternative fuels. The intermittent combustion process engines are somewhat more limited in flexibility, particularly the diesel engine which requires fuels of greatly restricted range in cetane number to sustain combustion in the compression ignition process.

An estimate of the current state of development and the overall required development time for each alternative engine is important in assessing the period when such an engine might be available in large quantities as a marketable automobile power plant. On the basis of information assembled in a study of gas turbine engine production implementation prepared for the U.S. Department of Transportation, Figure S-19 shows the estimated product development time for a new automobile engine. By assuming similar development time for various types of engines, one estimate of the current point of progress for each of the alternative engines under discussion has been recorded on the time scale. It should be noted that the pace of progress from this time onward is not necessarily the same for each engine. It is dependent on the assigned development funding levels and on the complexity of problems encountered in efforts made to meet engine performance and manufacturing design requirements. The Stirling engine lags behind all other engines and, were it to be mass produced, it is estimated to be almost nine years from full[*] production.

The rotary engine is only in limited[**] production at this time by Mazda. (General Motors plans to introduce this engine in limited quantities for the 1975 model year Vega.)

Diesels are shown as being in limited production. Mercedes Benz has a low performance diesel in production, and Peugeot has started production in model year 1975. However, a lightweight, high-performance engine for automobiles is still in a research prototype development phase and is estimated to be about 7 years away from full production.

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[*] Approximately 300,000+ units per year

[**] Approximately 20,000 units per year

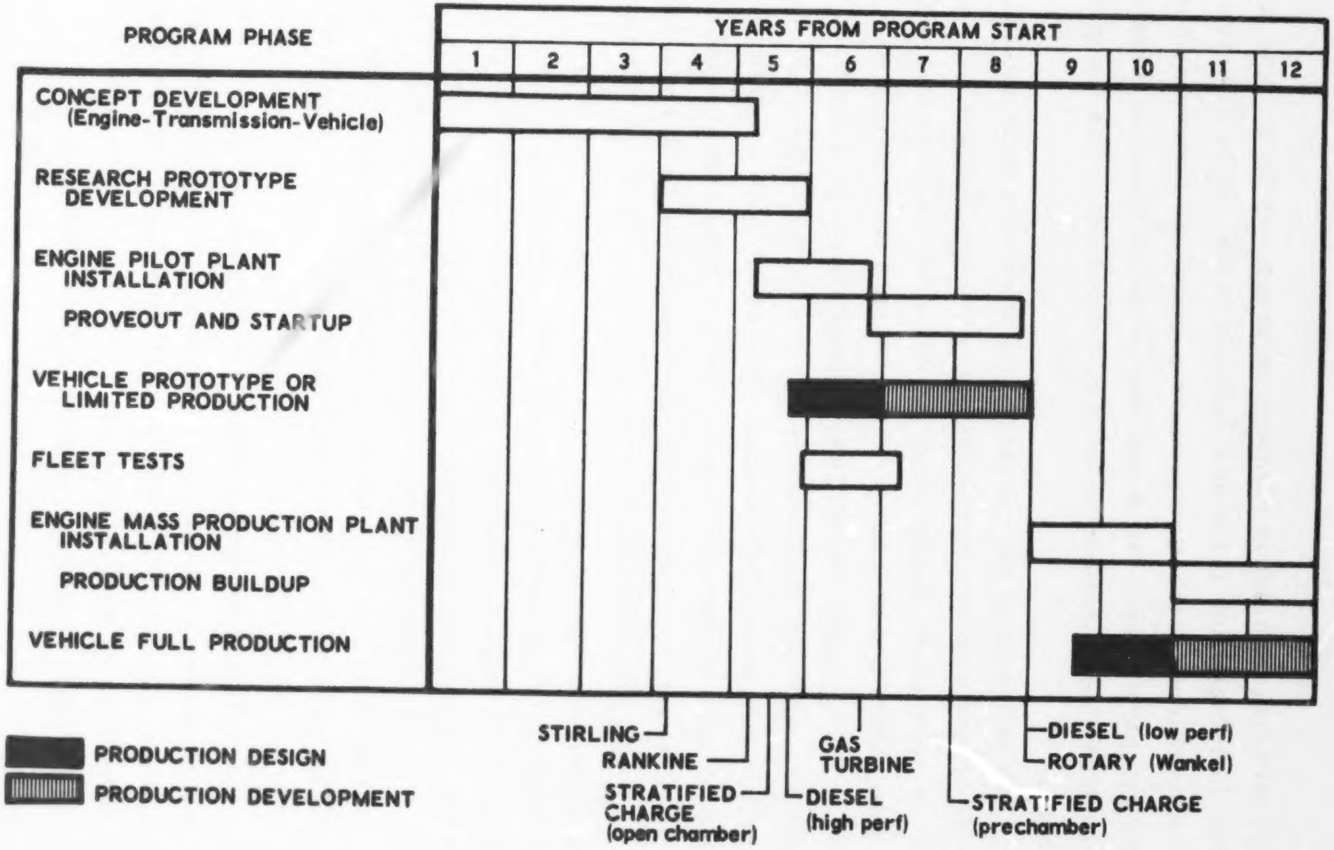

Figure S-19. Automobile Mass Production Implementation Schedule for a New Engine

The prechamber version of the stratified charge engine is considered to be in a limited production phase on the basis of Honda's introduction of their CVCC engine in the 1975 model year Civic. But Ford feels that their open-chamber design could only be produced in limited numbers by 1977.

If gas turbines were to be mass produced, it is estimated that these engines are about 6-1/2 years away from full production, followed by Rankine cycle engines at almost 8 years from full production. The proximity in time of the gas turbine engine to pilot plant installation is based on the limited production of engines that Chrysler carried out during their 50-car program, the experience gained by General Motors in small production of turbines for trucks, and the experience of Solar and Ford in carrying out the production design effort for limited production of turbines for trucks.

At this time no alternative engine development appears to have progressed to a point where sufficient data are available to substantiate any claim of superiority over other engines. Although the engines discussed in this report had been formerly viewed with the prospect of meeting the original 1976 Federal emission standards, the recent emphasis on conservation of natural resources has prompted a review of engine performance to promote high fuel economy and wide fuel adaptability in concert with low exhaust emissions. This trend may be seen in the renewed interest in the Stirling engine, whereby the Philips and United Stirling lightweight designs are being actively investigated by Ford for prototype installation in Torino and Pinto cars, respectively.

1. INTRODUCTION TO HEAT ENGINE SECTION

Enactment of the Clean Air Act Amendments of 1970 triggered vigorous research efforts on the part of the automotive and related industries to find the means to meet the original(*, **) 1976 exhaust-emission standards for hydrocarbon (HC), carbon monoxide (CO), and nitrogen oxides (NO_x) (0.41, 3.4, and 0.40 gm/mi, respectively) required for the 1976 model year automobile and beyond. Most of this industry effort was focused on attempts to clean up the conventional spark ignition internal combustion engine (ICE). Some of the research money went into investigating the feasibility of low-emission types of engines as an alternative to the conventional ICE.

The ability of the conventional ICE to meet the 1976-1977 Federal exhaust emission standards has been in doubt for some years. Even with the most elaborate emission control schemes, the prospects for meeting these standards are considered marginal by most auto makers. In addition, the add-on emission control systems are expensive, they affect drivability of the car, their durability for 50,000 miles is a serious problem, and they result in degraded fuel economy.

As an alternative, a number of other engines have been considered for the automobile power plant of the future. These engines, as conceived, may not require add-on devices. In most cases, it is expected that the basic combustion process can be adequately controlled to achieve the expected low emission levels. In this regard, the most promising

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[*] Interim standards with relaxed requirements are now in effect.

[**] Exhaust emission goals for engines under investigation in the AAPS Program are set at one-half the Federal standard. Requirements are made more severe to: (a) account for changes from prototype to mass-production designs, (b) ensure that production engines, with expected statistical variations, can pass the Federal certification test, and (c) allow for expected increases in emissions as vehicle mileage reaches the 50,000-mile Federal certification limit.

systems rely on continuous combustion processes as opposed to the intermittent combustion processes found in a spark ignition engine. However, low exhaust emission alone cannot be the sole basis for selecting an engine of the future. These advanced engines, when ranked against equivalent horsepower spark ignition engines, must also fulfill a number of other requirements, among which are:

a. Competitive fuel economy both in stop-and-go traffic as well as at highway cruise speeds

b. Physical dimensions compatible with automobile engine compartments

c. Ability to operate satisfactorily under a wide range of climatic and altitude conditions

d. Competitive first cost and maintenance cost

e. Adaptability to mass production manufacturing processes

f. Competitive durability

g. Adequate responsiveness to driver commands.

Evaluation of low-emission alternative engines to meet these requirements has been a continuing task since 1970 when a program of research and technology development was launched by EPA to evaluate the feasibility of various automotive power plants. The program is called the Alternative Automotive Power Systems (AAPS) Program and is administered by the Alternative Automotive Power Systems Division of EPA. This program was intended to supplement the work of industry, thereby ensuring that all reasonable technical approaches to meeting the standards were being pursued. A few of the engines under study were invented several decades ago and have been used successfully as power generators for a number of years. However, until the recent application of modern-day technology, these concepts were considered too inadequate in weight, size, cost, and efficiency for serious consideration as a mass-produced automobile power plant.

In July 1971 it was decided to limit the number of candidate systems under development and to concentrate all available funding and manpower resources on the three most promising systems for the near term.

These systems are the gas turbine, Rankine cycle, and stratified charge engines. The theoretical characteristics and anticipated development schedules of these three systems were considered more favorable than the others. Consequently, development work is focusing on these three, with major efforts going to the gas turbine and Rankine cycle engines because of their greater need for new technological advancements.

By means of hardware demonstration of alternative power systems, it is hoped that the AAPS Program will stimulate industry to absorb technologies developed in the program and further develop the candidate systems into production engines. Since its inception it has been apparent that the program has served additionally as an important mechanism for transferring aerospace technologies developed under federal government funding in NASA and DOD programs.

This volume of the report to the Administrator is intended to provide a summary of the status of engine development work in this program as well as in similar alternative engine hardware development programs being funded by private industry. Operating characteristics of the alternative systems are also provided. Still more information on other developments is known to the government but cannot be documented herein because the data are proprietary. Finally, the status of developments as of the end of 1973 is summarized, wherever possible, in the report.

One purpose of the AAPS Program serves to provide a basis for knowing what can and cannot be accomplished with alternative engines used in practical ways; thus, the publication of such information as contained herein is in keeping with the mission of the AAPS Program mission. It is intended that this series of reports on alternative engines be published annually, with the present report being the first. The program is scheduled to be completed in 1975 with demonstrations, testing, and evaluations of Rankine cycle and gas turbine engines installed in automobiles.

2. OPERATIONAL REQUIREMENTS AND CHARACTERISTICS OF AUTOMOBILES POWERED BY CONVENTIONAL SPARK IGNITION ENGINES

This section of the report briefly addresses and summarizes the salient features and characteristics of automobiles powered with conventional spark ignition engines. It is included herein in order to provide a basis for perspective and comparison between the conventional spark ignition engine and the various alternative automotive engines examined in detail in later sections of this volume--in particular, with regard to vehicle performance, fuel economy, and exhaust emissions.

Initially, a brief historical review of the evolution of the spark ignition engine-powered automobile is presented in Section 2.1. Next, automobile systems and operational requirements are discussed with respect to the interrelationships influencing vehicle performance capability and fuel economy (Section 2.2). Finally, Section 2.3 summarizes exhaust emission control systems applicable to the conventional spark ignition engine and resultant exhaust emissions as characterized by current federal exhaust emission standards. Appendix A contains a listing of contemporary vehicle design specifications as provided by the AAPS Program to form a reference vehicle for comparisons of alternative engine performance when installed in the automobile.

2.1 EVOLUTION OF THE SPARK IGNITION ENGINE*

2.1.1 Early History of Auto Engines

The task of determining why the internal combustion is the engine of primary use is as difficult as determining who discovered America.

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*Selected abstracts from Reference 2-1.

The historical record of locomotive power is an exercise in love and confusion. The affection of authors and historians for one or another of the electric, steam, or internal combustion engines results in the favored machine being bathed in rose-colored lights. Confusion exists because few sources -- including the papers by the principals themselves -- agree even on dates, let alone on the factors that influenced the ultimate success or failure of the power unit.

Nevertheless there are numerous identifiable factors, individuals, and circumstances involved in the trail that leads ultimately to our current automobile. Among these are the political, sociological, economic, technological (internal and external), and institutional forces that influenced the decisions of individuals and firms. Personal drives of engineers, promoters, and financiers were also instrumental in shaping the current automobile industry.

The critical period for the fledgling auto industry was from about 1900 to about 1920:* The period *** was *** one of ferment and growth for the entire automobile industry. That activity was in fact a phantasmagoria of bold ideas, ingenious and stubborn experimentation, enthusiastic promotion, dramatic accomplishment, and weird display. From it developed the steadier era of manufacturing in which Henry Ford would emerge as a leader.

Although by 1900 at least 57 American plants were engaged in making motor cars, many of which were in an experimental state, the basic type of vehicle had not yet been determined, and it was uncertain where the chief centers for the new industry would develop. This year saw about 4,000 American cars manufactured, of which steam and electric models made up more than three-fourths. One respectable set of figures lists 1,681 steam carriages, 1,575 electric, and only 936 gasoline cars.

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*Abstracted from Reference 2-2.

2.1.2 Factors Affecting Early Decisions in Favor of Spark Ignition Engines

The literature of the late 19th and early 20th centuries, as well as more recent studies of that literature and of that period, does not provide the answer to the question as to why the spark ignition engine became the engine of choice. However, that literature does support the following conclusions:

a. At the time of its original selection as the engine of choice, the spark ignition engine had no overall intrinsic advantage over other types of engines.

b. The spark ignition engine became the engine of choice as the result of a number of circumstances and events rather than as the result of a formal decision-making process. Among the circumstances and events were:

    1. Although steam-powered road vehicles were developed in the 1820s, restrictive legislation in England in force between the 1830s and 1896 discouraged further developmental efforts during the period of conception and early development of the spark ignition engine.

    2. Henry Ford picked the spark ignition engine to power his cars. When the Model T succeeded, the spark ignition engine was assured of at least a place in the range of auto engines to be used. Why Ford picked the spark ignition engine is not clear, beyond his great concern for light weight.

    3. Once Ford was in mass production, his competitors had to pick a basis on which to compete against him. His most successful competitor, General Motors, decided to compete on the basis of financial and marketing values and not on the basis of engine type.

    4. Standard Oil Company, with its development of the Burton process to crack crude oil, assured a plentiful supply of inexpensive, though "highly refined," gasoline needed by the spark ignition engine but by none of the other types.

    5. Adding the electric starter, which removed the problem (and danger) of hand-cranking to start the engine, opened the women's market to spark ignition-powered cars. Up to that point, women had preferred the electric car.

6. When Doble entered into competition against Ford with a steam-powered car, World War I came along, causing the Government to intervene in the interest of war production. Doble did not fill $27 million in standing orders; when the war was over any momentum that might have built up for Doble's steam-powered competitor was gone forever.

7. In the early 1920s, when roads were becoming good enough to support a demand for higher engine performance, Midgley discovered tetraethyl-lead which, when added to gasoline, provided the spark ignition engine with the performance capability to defend its position against the known high-speed performance of the steam-powered engine.

c. As car buyers increased the stringency of their demands for engine performance, the industry successfully modified the spark ignition engine to satisfy those demands. There has been no Achilles heel in its performance on which competitive types of engines could base a challenge.

2.1.3 Factors Influencing Continued Reliance on Spark Ignition Engines

Fierce arguments raged for a quarter century or more over whether the spark ignition engine, the steam engine, or the electric battery would be the best way to power a car. Reading those arguments today, one cannot help but wonder at the willingness of any entrepreneur to risk his all on any one of them. But at least he didn't have to worry about displacing an engine already performing satisfactorily, as becomes apparent when the following factors are considered:

a. In contrast to the situation in 1900, the spark ignition engine today has a fundamental intrinsic advantage over all other candidates. Over more than 50 years, hundreds of millions of them have been made, used, maintained, and repaired:

1. Massive production facilities, tools, and methods exist for their large-scale production.

2. Designers have an immensely detailed and documented understanding of factors affecting their design, performance, and cost.

3. Hundreds of thousands of people are trained in their maintenance and repair.

4. A network of service facilities, with tools and spare parts distribution, has been built up.

5. Millions of customers through personal experience have gained confidence in spark ignition engines.

6. There is incontrovertible evidence that progress is currently being made in designing them to operate at lower emission levels without intolerable sacrifices in other parameters of performance.

b. There are no equivalents in knowledge, experience, or data with alternative engine designs. Thus it is not possible to prove that a change to one of these alternative designs would be beneficial -- in either emission reduction or other performance parameters.

c. Reducing emissions is not the only change in engine design parameters that must be incorporated over the next decade. Fuel source, fuel economy, and transportation functions are in process of reconsideration as a result of major shifts in population, social values, and resources. Many of the factors involved in these changing parameters are not yet clear. In light of such uncertainty, decision makers seek to keep open as many options as possible for as long a time as possible or, conversely, to make short-term decisions as incremental as possible and to postpone major "change" decisions.

d. All technological innovation, including incremental steps, follows a pattern that can be expected to apply to the auto industry. Typically, an innovation is introduced on a small scale, tested, and proved; gradually it penetrates the market. The period of experimentation varies widely, depending on numerous factors, including those listed above in a and b, and involving the complexity of the innovation, the extent of the supporting system for the existing technology, and the social values affected. Several years is almost certainly the shortest period in which a major innovation can fill a market opportunity.

While the process of innovation can be shaped and directed to some extent by legislation, research and development, and investment, there are limits to how much it can be affected.

The issue is not only whether the evaluation period can be shortened, but also -- in the light of the long-term changes in the offing, as noted in c, and the reductions in emissions from spark ignition engines currently being achieved -- whether the evaluation period should be shortened.

2.2 AUTOMOBILE SYSTEMS AND OPERATIONAL REQUIREMENTS: INTERRELATIONSHIPS INFLUENCING PERFORMANCE AND FUEL ECONOMY*

The performance and fuel economy of an automobile is determined by detailed engineering characteristics of the specific vehicle, vehicle operating requirements, and demands of the driver. The following provides a brief explanation of some of the basic interrelationahips of design and operating parameters, providing a generalized background for the interpretation of detailed evaluations of changes in vehicle design characteristics.

The basic physical law governing the motion of an automobile is Newton's second law of motion. In order to place an automobile in motion a force must be exerted through the vehicle driving wheels. The amount of the force depends on the weight of the car and the rate of vehicle acceleration, as well as the magnitude of forces which retard motion such as rolling resistance and aerodynamic drag.

The motive force is applied to the vehicle driving wheels by a combination of the operation of the vehicle engine, transmission, and rear axle. The function of the transmission and rear axle is to deliver the turning effort of the engine to the driving wheels at a speed different from that of the engine. This difference in speed also implies a multiplication of the turning effort of the engine, depending on the value of the overall gear ratio.

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*This section based principally on selected abstracts from Ref. 2-3.

The turning effort of the engine is termed torque and arises as a result of ignition and subsequent expansion of the products of combustion of a hydrocarbon fuel and air mixture within the cylinders of an internal combustion engine. Detail design characteristics of specific internal combustion engines determine the amount of torque produced by that engine. Basically, however, it is the quantity of fuel and air ingested by the engine during each revolution that determines the amount of torque produced. Typically, small-displacement engines produce less torque than larger-displacement engines, all other factors being equal.

Traffic conditions require that a vehicle be operated at varied speeds and accelerations, all of which require varied torques. Ignoring the transmission momentarily, the torque needed to sustain the desired operating condition is obtained by varying the amount of fuel air mixture processed by the engine. For a vehicle with a carbureted engine this is, in effect, accomplished by the modulation of the accelerator pedal. Depressing the accelerator allows more mixture to enter the engine, thus producing more torque. Of course, there is a limit to the amount of torque that any engine with fixed design characteristics can produce. This maximum torque at any given engine speed is produced at wide open throttle (WOT). The generalized torque characteristic of an internal combustion engine is shown in Figure 2-1.

The transmission is used to multiply the torque output of the engine so that the vehicle can be subjected to motive forces higher than could be developed by the direct connection of the engine to the rear axle. Transmission gear ratios greater than 1:1 allow the torque necessary to move the vehicle to be delivered at smaller throttle (accelerator) settings. To meet any given vehicle torque requirement, many gear ratios and various combinations of engine displacement and throttle openings can be used.

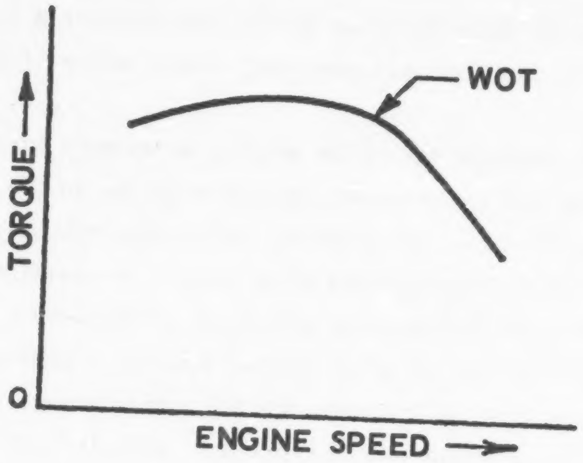

Figure 2-1. Generalized Engine Torque Characteristic

Another basic concept which must be considered in the discussion of automobile performance and economy is power. To define what is meant by power we must go back to our explanation of the force acting at the wheel of the vehicle. As the force acts on the vehicle, the vehicle begins to move. The force, of course, then acts on the vehicle over the distance through which the vehicle moves. This quantity (force × distance) is termed work, and the rate at which the work is done is termed power.

In engineering terms, the power output of an engine is specified as Power = Torque × rpm Constant. The constant is selected to get the units of power into a comparative unit of measure, such as horsepower. In essence, power is the price one has to pay when demanding a given force at a given speed. It is the power requirement of the desired motion of an automobile, relative to the overall efficiency of the translation of the energy of combustion into meeting this power demand, that determines automobile economy.

Before further deliberations on the fuel economy of automobiles, it is necessary to note that other physical laws are also of primary significance to the subject of fuel economy. The field of thermodynamics provides guidance in this area and establishes theoretical efficiency boundaries below which real systems can operate. In particular, no practical internal combustion engine can have a thermal efficiency (power output/power available from fuel combustion) over about 50 percent; i.e., it is not possible to use more than 50 percent (and usually considerably less) of the energy available in the fuel. The remainder is rejected to engine exhaust and cooling systems. Detail design characteristics and operation of various engines dictate how close to this theoretical limit real engines can operate. An example of estimates of thermal efficiency for various engines powering an automobile over the Federal Driving Cycle is given in Figure 2-2. Note that the efficiency of a number of the alternative engines equals or exceeds that for the spark ignition engine.

The conventional naturally aspirated (NA) carbureted engine demonstrates a performance (torque vs rpm) map under fully warmed-up conditions similar to Figure 2-3. The contours are lines of constant fuel consumption for each unit of horsepower developed (brake specific fuel consumption). This characteristic map results from two primary effects. The first is the increase in brake specific fuel consumption (BSFC) with decreasing torque output at a constant engine speed. This is due to the higher percentage of friction power at low brake power outputs. Second, brake specific fuel consumption increases with increasing speed (at constant torque) due to increased engine friction. Consequently, the highest fuel use efficiency for the engine occurs at low engine speeds and high torque outputs.

The engine performance map is an extremely significant constituent of the overall vehicle fuel economy formulation. This map is the representation of the units of fuel consumed in meeting the power demanded by the motion of the vehicle. The values of the brake specific fuel consumption, as well as the shape of the characteristic fuel-consumption curves are extremely significant in determining accurate estimates of economy.

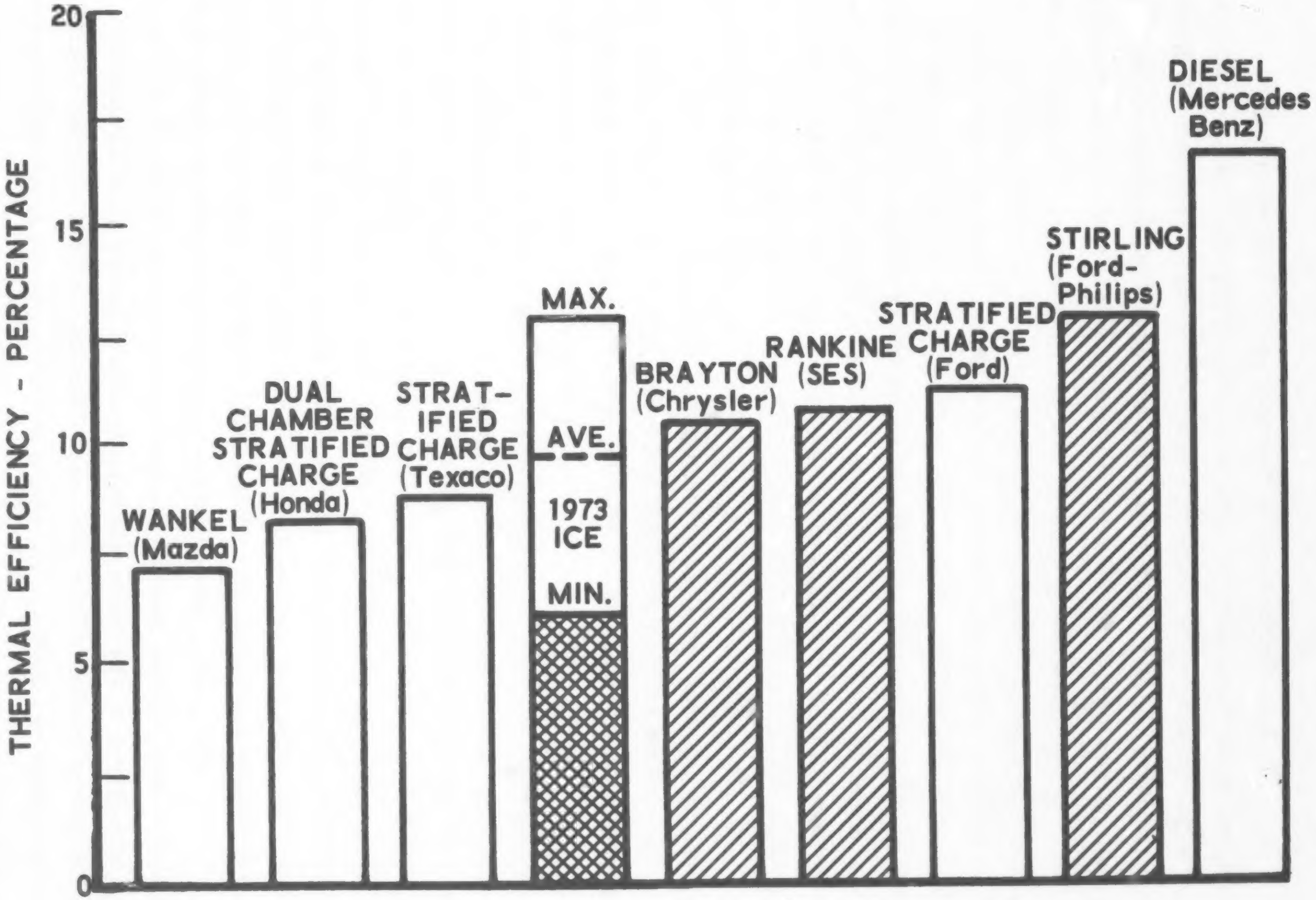

Figure 2-2. Thermal Efficiency over the Federal Driving Cycle (Ref. 2-4)

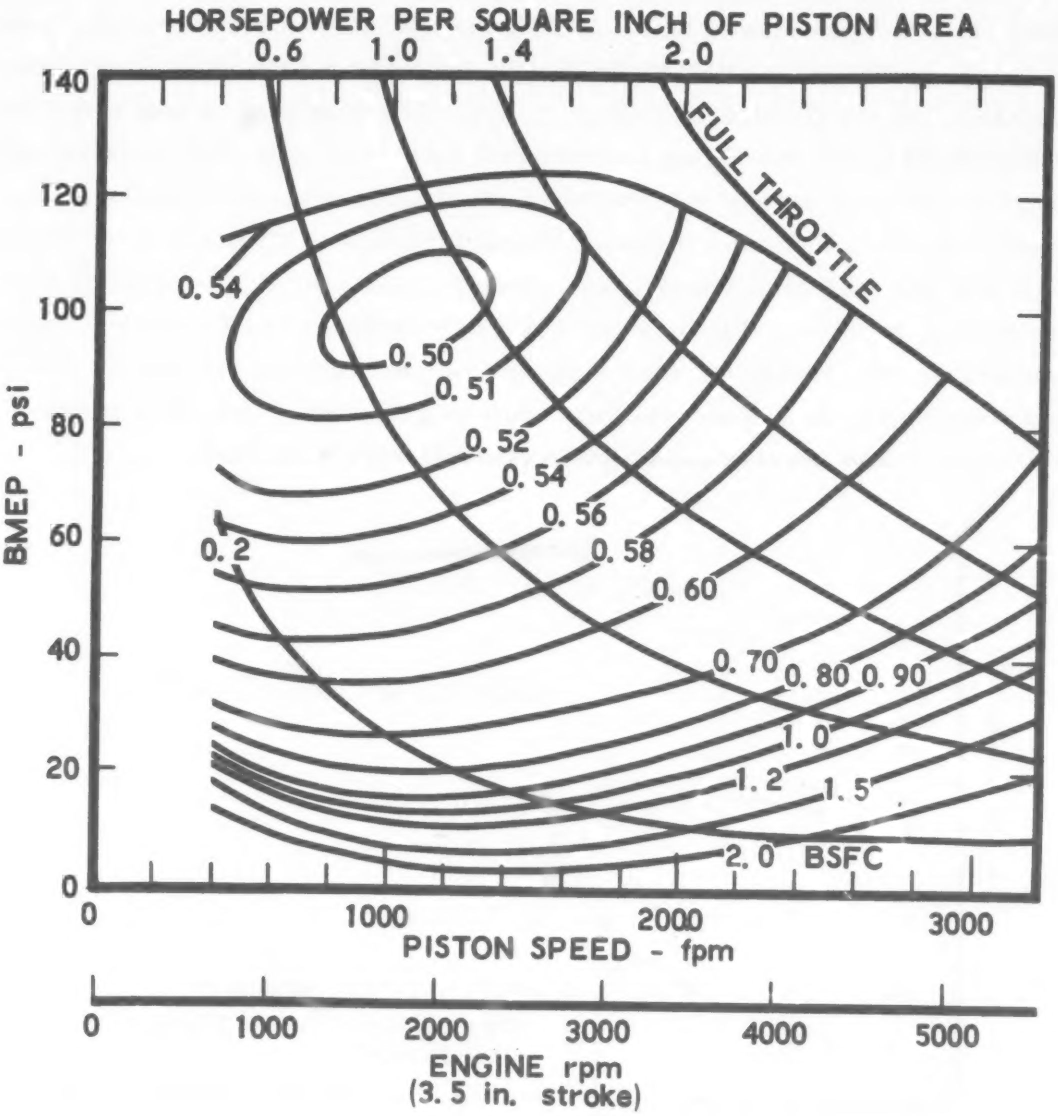

Figure 2-3. Performance Map of a Carbureted Spark Ignition Engine (Ref. 2-5)

Unfortunately conventional applications of engines to vehicles result in a compromise of where the engine operates in the map under road-cruising conditions relative to the maximum performance capability of the vehicle. This is illustrated in Figure 2-4. The road load is met at points of high BSFC, and the difference between the torque capability at full throttle and the road-load torque value determines the performance (acceleration) capability of the vehicle. If design characteristics of a vehicle were adjusted such that the road-load power requirements could be met by operating near the point of minimum BSFC, a marked improvement in fuel economy could then be gained. However, such a move with conventional technology (such as a very small engine) would also result in substantial reduction in the acceleration and passing capabilities of a full-size automobile.

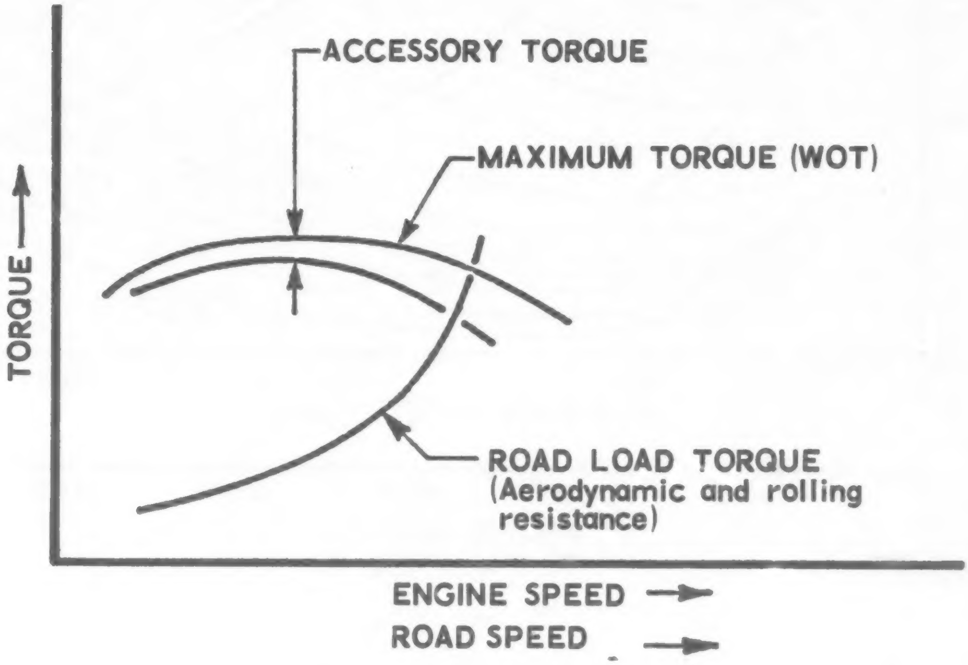

Figure 2-4. Generalized Engine and Vehicle Torque Characteristics

The foregoing discussion stressed the difference between the full power capability of a vehicle and the power necessary to maintain level-road motion (no acceleration). For a full-size automobile, typical road-load power requirements are given in Table 2-1.

Table 2-1. Typical Load-Power Requirements (Ref. 2-3)

Speed (mph)	Power (hp)
20	5.0
30	8.2
40	13.0
50	19.0
60	28.0
70	38.5

This power requirement is determined by a large number of factors including vehicle weight, tire design and construction, road surface conditions, air temperature and density, and vehicle size and shape. The vehicle design must accommodate a range of power requirements that result from operation throughout the United States. For example, at an altitude of 4,000 feet, the road load at 70 mph is reduced by 3.8 horsepower, or about 10 percent, as a result of the decreased air density. Operating a vehicle in a 0°F ambient instead of a 100°F ambient increases the road load by about 12 percent at 70 mph. Fuel economy, then, can be seen to be influenced by ambient conditions as well as the vehicle operating requirements.

The fuel economy measured under level-road conditions is one method for comparing vehicle designs; however, the vehicle must be capable of meeting varied road conditions, such as grades. "Level" sections of interstate highways aren't always level. Often highways include grades, and a 3-foot rise in 100 feet of road is not uncommon. The conventional American automobile can negotiate such grades without requiring downshift to a lower gear. The significance of grades on vehicle-power requirements is illustrated by Table 2-2.

Table 2-2. Three-Percent Grade Power Requirements for a 4,300-pound Vehicle (Ref. 2-3)

Speed (mph)	Additional Grade Horsepower	% Increase Over Level Road
20	6.88	137.0
30	10.32	143.0
40	13.75	105.0
50	17.19	90.0
60	20.63	74.0
70	24.07	63.0

Negotiating a grade of only 3 percent more than doubles the road-load power requirements at speeds up to 40 mph and also provides a substantial increase even at 70 mph.

A vehicle can't be designed to maximize economy under specific level road conditions but must be designed to accommodate a wide range of driving conditions encountered in the United States. In addition to steady-state driving conditions, transient operation of vehicles is also extremely important. Stop-and-go traffic conditions such as accelerations, braking, idling, etc., constitute typical driving conditions experienced by the driving public.

Under acceleration, weight is the single most important parameter influencing vehicle-fuel consumption. Figure 2-5 clearly establishes the magnitude of this power demand for cars powered by spark ignition engines. Under the maximum acceleration requirements of the

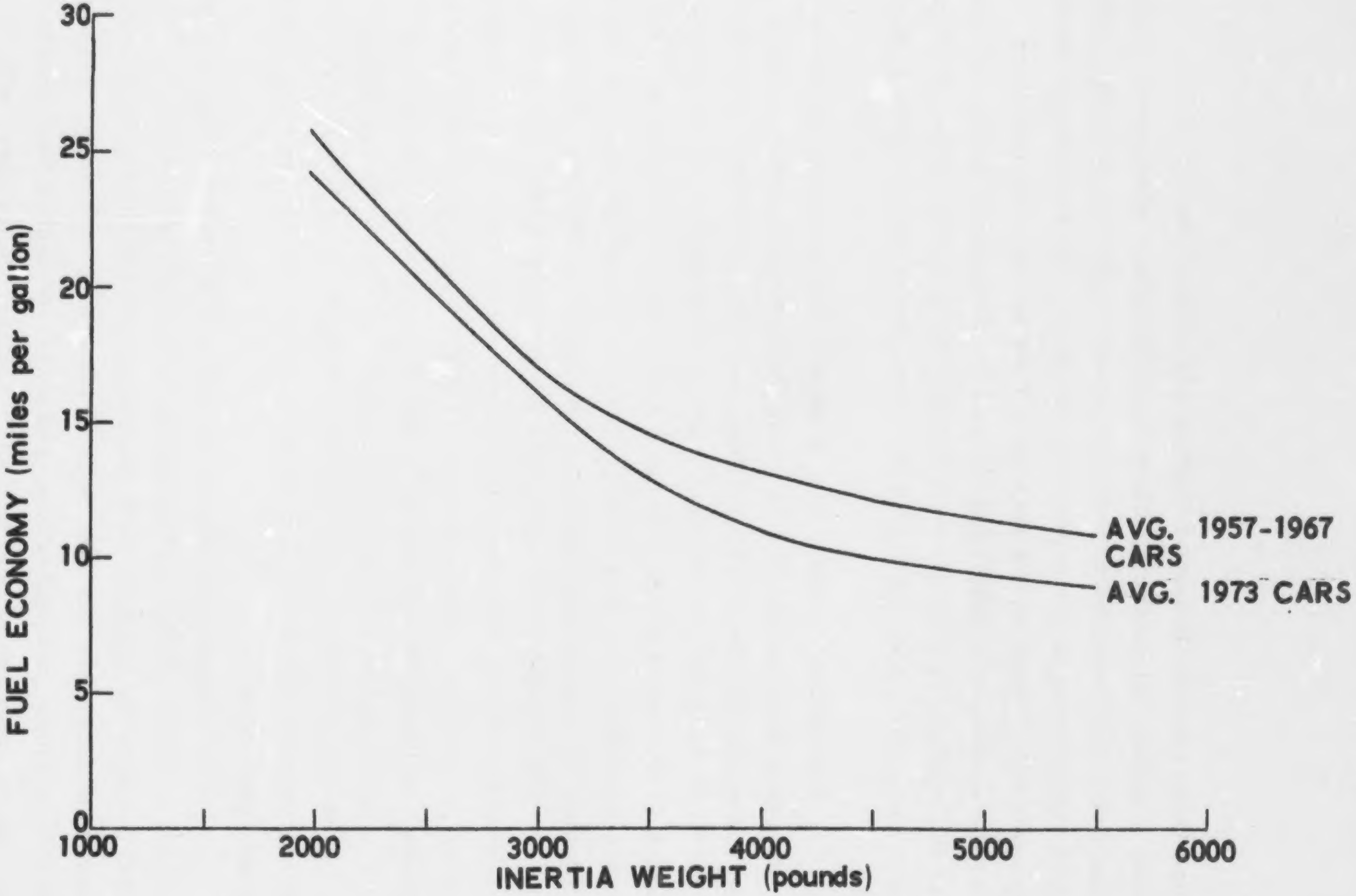

Figure 2-5. Fuel Economy over the Federal Driving Cycle (Ref. 2-4)

EPA Urban Dynamometer Driving Schedule (also called the LA-4 dynamometer cycle), the power required to accelerate a 4300-pound vehicle is over six times that required to maintain steady motion at 30 mph. (The influence of vehicle weight can be marked; however, the significance over a driving cycle will depend on the amount of time spent in acceleration modes and the amount of acceleration demand.) Of concern to the motoring public and the engine designer is the noticeable drop in fuel economy of the later model cars that come equipped with more elaborate emission control equipment.

For further comparison, Figure 2-6 illustrates the power requirements of a 4300-pound vehicle under the same acceleration loads and the power requirements of accelerating the vehicle at other performance levels. It should be noted that the maximum performance level of a full-size automobile lies somewhere between 5 and 6 seconds to 30 mph. The performance capability of a given vehicle weight then can be seen to significantly influence the power required and, consequently, the fuel consumed.

The weight of the vehicle is the primary factor that determines the installed power requirements of the vehicle when a given level of performance is specified. For example, in determining the power necessary to accelerate a 4300-pound vehicle to 30 mph in 5 seconds, less than a 10 percent error would be encountered in totally ignoring the road load. Conversely, unless the performance level or weight of a full-size vehicle is significantly reduced, the maximum power output of the engine must remain at previously used levels.

The influence on vehicle operating fuel economy due to starting and operating a vehicle "cold" as compared to "warm" is significant. The loss in fuel economy is attributable to varied sources including carburetion enrichment to maintain the air fuel ratio of the engine cylinders, viscous losses in cold engines, transmissions, and axles and higher heat losses in the engine due to rejection of heat to "cold" coolant. Since these effects are thermal, ambient temperature would be expected to influence the amount of

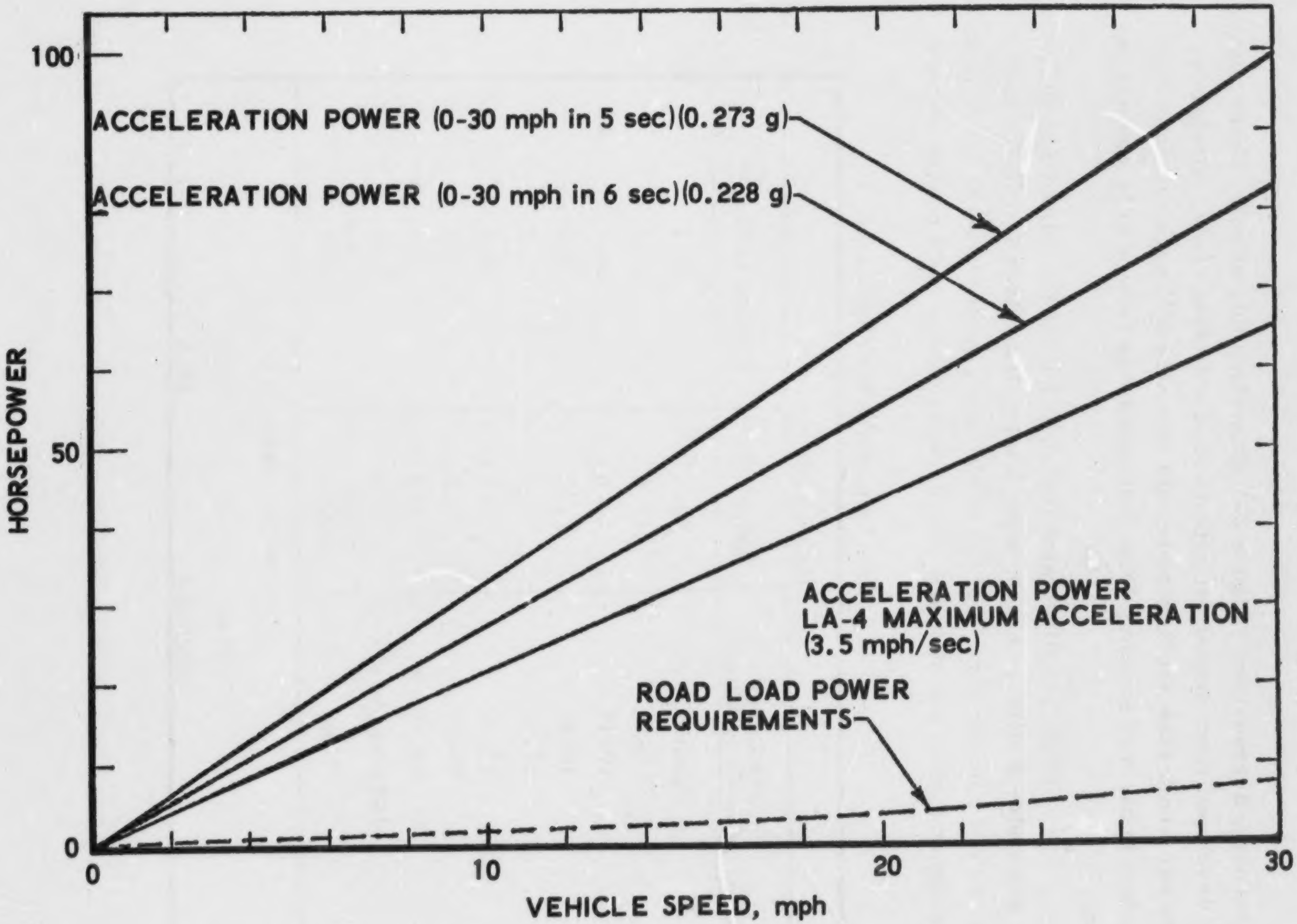

Figure 2-6. Motive Power Requirements, 4300-Pound Vehicle (Ref. 2-3)

fuel economy degradation. Figure 2-7 illustrates this effect. These curves were developed from sequential repetition of a driving cycle. Obviously, the driver who makes short trips in cold climates will suffer a significant reduction in his fuel economy from that obtainable from a fully warmed-up engine.

Table 2-3 illustrates that over half of the trips taken in the U.S. are under 5 miles. Although the total miles driven for these short trips are not a major portion of the total travel, improvements in the fuel consumption during warmup will also decrease the overall energy demand.

Table 2-3. Auto Trip Statistics (Ref. 2-6)

Urban Mileage		
Trip Length (One-Way Miles)	Trips, %	Vehicle Miles, %
Under 5	54.1	11.1
5-9	19.6	13.8
10-15	13.8	18.7
16-20	4.3	9.1
21-30	4.0	11.8
31-40	1.6	6.6
41-50	0.8	4.3
51-99	1.0	7.6
100 and over	0.8	17.0
Total	100.0	100.0

Total Mileage, %	
Urban	55.5
Highway	44.5

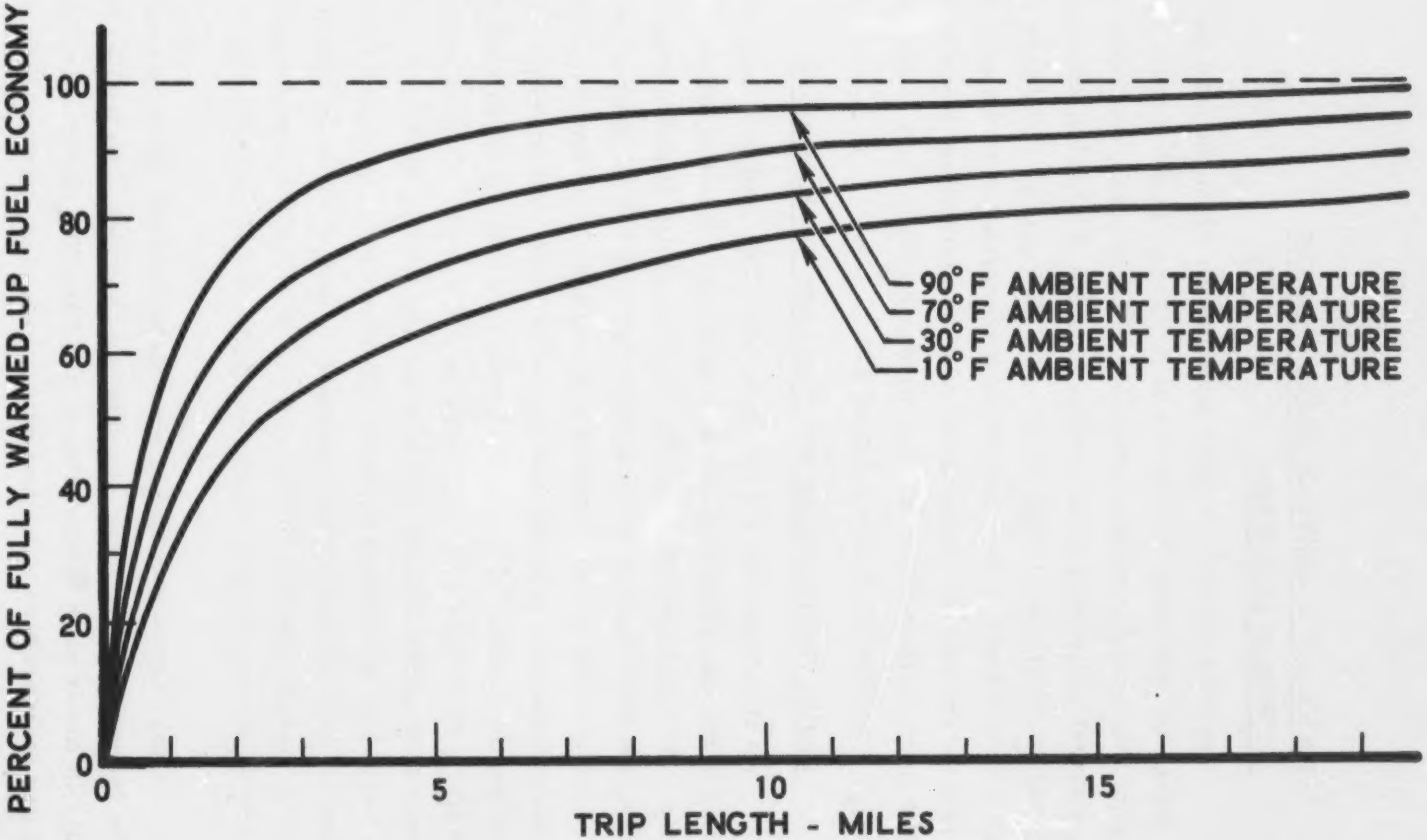

Figure 2-7. Warmup Economy (Ref. 2-7)

2.3 EXHAUST EMISSIONS AND EMISSION CONTROL SYSTEMS

Gaseous emissions from automobile exhausts may be controlled either by inhibiting formation of the gases in the engine cylinders or by lowering their concentration externally. In general, methods of controlling exhaust emissions from automotive spark ignition internal combustion engines to meet the stringent 1975-76-77 Federal standards involve certain engine modifications and the use of a combination of several devices. Multiple methods are necessary because of the requirement for simultaneous control of the hydrocarbon (HC), carbon monoxide (CO), and oxides of nitrogen (NO_x) constituents in the exhaust gas.

2.3.1 Engine Modifications and Operating Considerations

As illustrated in Figure 2-8, HC, CO, and NO_x concentrations in the exhaust of uncontrolled engines are strongly a function of the operating air fuel ratio. As can be noted from the figure, at the stoichiometric air fuel ratio NO_x production is very high while HC and CO production is relatively low. For air fuel ratios between approximately 17 and 19, levels for all three constituents are reduced considerably from peak values. Currently, engine operation is restricted to air fuel ratios below approximately 19 to avoid excessive power loss and rough engine operation. Operation in the 17 to 19 range minimizes HC and CO levels, and lowers that of NO_x, but the concurrent reduction of exhaust levels of all three species is far from sufficient to meet 1975-76 emission requirements. NO_x formation can be suppressed by operating in the "rich" regime (air fuel ratios of approximately 11-13); however, in this region HC and CO concentrations are very high.

Other factors affecting emissions include spark timing and design of the induction system and combustion chamber. Retarding the spark results in lower peak temperatures and less NO_x formation. Also, the exhaust gas temperature is higher with a retarded spark, which promotes

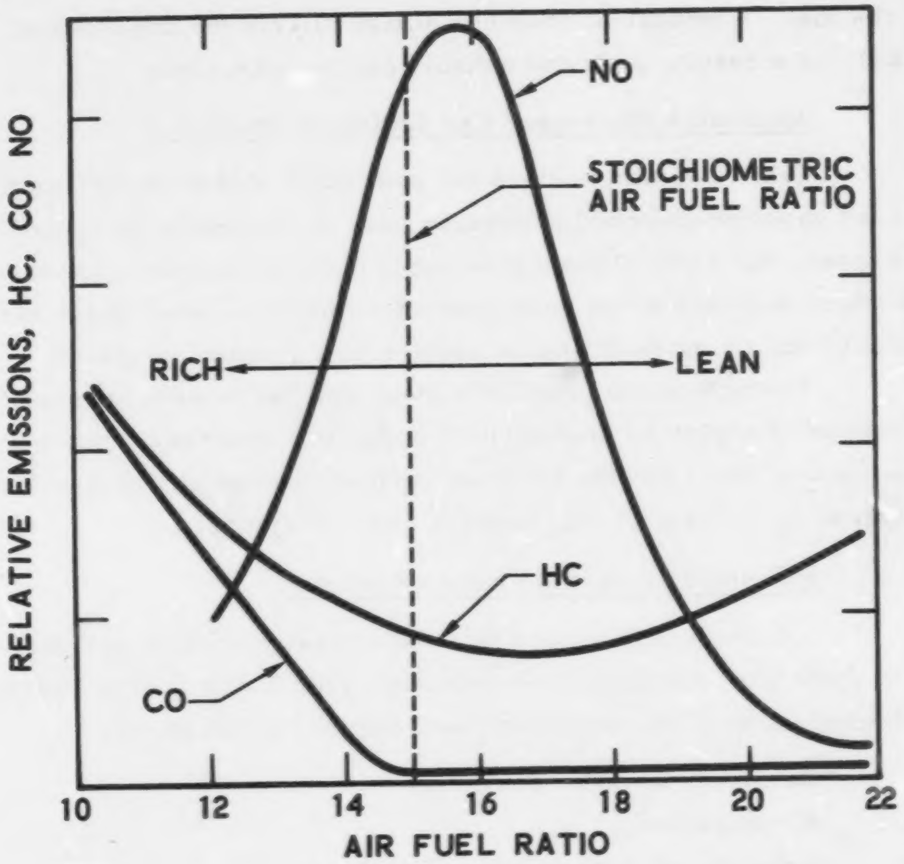

Figure 2-8. Effect of Air Fuel Ratio on Emission Levels, Gasoline Spark Ignition Engine (Ref. 2-8)

further combustion of the HC and CO species in the exhaust system. Induction-system modifications can result in lower emissions by providing a more uniform mixture to the cylinders and better atomization and vaporization of the fuel. Combustion chamber design affects the combustion process and, as a result, peak and exhaust gas temperatures.

2.3.2 Applicable Passenger Car Emission Standards

Table 2-4 summarizes the applicable statutory emission standards for gasoline-powered passenger cars as currently promulgated. As can be noted, the Federal standards apply nationwide except for those instances where California has been granted a waiver to adopt more stringent control of one or more emission species in a given time period.

Through model year 1974 these standards have been met by the combination of engine modifications (combustion chamber redesign, spark retardation, etc.) and the incorporation of exhaust gas recirculation (EGR) systems for control of NO_x to meet 1973-74 standards.

2.3.3 Projected Emission Control Systems

It is generally agreed that the emission control system required to meet the California 1975 standards (see Table 2-4) is exemplified by the following package of components and engine modifications:

Oxidation catalytic converter
Air injection
Exhaust gas recirculation (EGR)
Carburetor modifications
Ignition system modifications.

There is not complete agreement as to whether the catalytic converter will be required to meet the less stringent Federal 1975 standards that apply to the rest of the nation. However, some manufacturers have implied that at least some of their car models sold in states other than California will also

Table 2-4. Summary of Passenger Car Emission Standards (Gasoline Powered)

Applicable Year and Area		Emission Rate, Grams/Mile																
		Uncontrolled Pre-1966[a]		1966 Cal 1968 Fed	1970		1971		1972		1973		1974		1975[c]		1976[d]	1977[d]
					Cal	Fed	Cal	Fed	Cal	Fed	Cal	Fed	Cal	Fed	Cal	Fed	Nation	Nation
Test Procedure		FTP	CVS-C	FTP (Concentration Measure)					CVS-C						CVS-CH			
Emission	Hydrocarbons	10	17	3.4	2.2	2.2	2.2	2.2	3.2	3.4	3.2	3.4	3.2	3.4	0.9	1.5	0.41	0.41
	Carbon Monoxide	77	125	35	23	23	23	23	39	39	39	39	39	39	9.0	15	3.4	3.4
	Nitrogen Oxides	4-6	4-6	NR	NR	NR	4	NR	3.2[b]	NR	3.0	3.0	2.0	3.0	2.0	3.1	2.0	0.4
	Evaporative (grams per test)	40		NR	6	NR	6		2		2		2		2		2	2

FTP	7-mode Federal test procedure
NR	No requirement
CVS-C	CVS cold start mass test
CVS-CH	CVS cold/hot weighting mass test

[a] Estimate only, for comparison purposes

[b] Two hot cycles of FTP test

[c] Original 1975 standards were: HC = 0.41, CO = 3.4, and NO_x = 3.1

[d] Original 1976 standards were same as current 1977 standards

incorporate the catalytic converter. An example of emission control system elements for control of both exhaust and evaporative emissions is given in Figure 2-9 for General Motors' projected 1975 design.

The oxidation catalyst with air injection is provided to promote the oxidation of the unburned hydrocarbon (HC) and carbon monoxide (CO) species contained in the engine exhaust. The catalyst type which appears most frequently among the selected first-choice systems is the platinum-group metal/monolithic converter. Base metal/pelletized and promoted base metal/pelletized catalyst designs* also are being evaluated by some manufacturers. Some automobile manufacturers are also considering the utilization of a catalyst overtemperature protection device to prevent catalyst damage under extreme or abnormal engine operating conditions (spark plug misfire, etc.).

Exhaust gas recirculation (EGR) systems for suppression of NO_x formation will be employed in nearly all of the 1975 model year automobiles. These systems will be improved versions of the EGR systems used in most of the 1973 and 1974 model year vehicles.

The emission control systems of a number of manufacturers utilize air injection in the exhaust manifold to provide more rapid warmup of the catalyst under cold-start conditions. A full-size thermal reactor is expected to be used in the 1975 system for the Mazda rotary engine.

Carburetion/intake system modifications range from complete redesigns utilizing new concepts to minor improvements to the current conventional systems. These modifications are generally directed toward improving the precision and stability of the air fuel ratio and also include such features as altitude compensation, quick-release choke devices, and intake-manifold heating.

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* Promoted base-metal catalyst formulations contain small amounts of platinum-group metals.

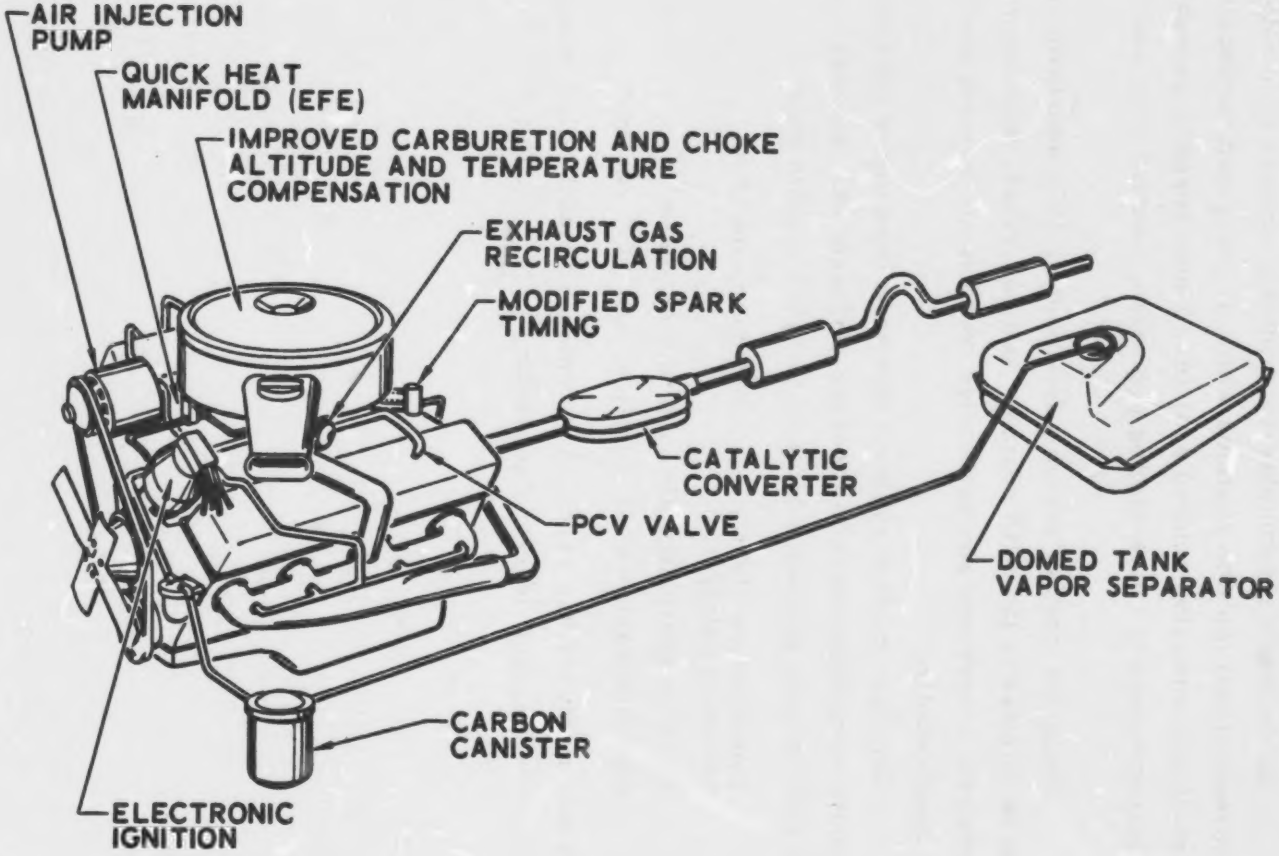
AIR INJECTION PUMP
QUICK HEAT MANIFOLD (EFE)
IMPROVED CARBURETION AND CHOKE ALTITUDE AND TEMPERATURE COMPENSATION
EXHAUST GAS RECIRCULATION
MODIFIED SPARK TIMING
CATALYTIC CONVERTER
PCV VALVE
DOMED TANK VAPOR SEPARATOR
CARBON CANISTER
ELECTRONIC IGNITION

Figure 2-9 . General Motors Projected 1975 Under-Floor Emission Control System (Ref. 2-9)

All domestic manufacturers propose, or have in development, electronic (breakerless) ignition systems which are targeted for inclusion in their first-choice emission control system. These systems generally provide an improvement in spark-timing precision, consistency, and reliability.

Emission control systems for meeting 1976 standards are projected to be similar to the 1975 systems just described, with improvements in catalytic converters and associated components to meet the lower HC and CO requirements.

Emission control systems that have been under consideration by the automobile manufacturers to meet the stringent NO_x standard prescribed for 1977 include all components of the 1976 system plus:

Reduction catalyst(s) installed upstream of the
oxidation catalyst(s)

More sophisticated air injection systems

Modified carburetion, ignition, and EGR systems.

To date, no manufacturer has reported acceptable durability for reduction catalysts, however further development testing is in progress.

3. GAS TURBINE ENGINES

3.1 INTRODUCTION

3.1.1 General Description

In its simplest form, the gas turbine engine consists of a compressor, a combustion chamber, and a turbine (see Figure 3-1). Air is taken in by the compressor at atmospheric pressure, compressed to a higher pressure, and is then delivered to the combustion chamber where fuel is injected and burned at essentially constant pressure. The resulting high-temperature, high-pressure gas is expanded in the turbine and then exhausted to the atmosphere. Part of the turbine-shaft work is used to drive the compressor; the remainder provides useful output work.

Several modifications of this simple cycle have been utilized in the development of gas turbine engines for vehicular use. Most engines currently in use for on-highway vehicles employ a free power turbine; i.e., two mechanically independent turbine stages are used (see Figure 3-2). One drives the compressor and the other drives the output shaft. This allows a wide range in output-shaft speed, which is not possible with a single-shaft turbine engine. To improve efficiency, such free turbine engines use a regenerative cycle. In a regenerative cycle, the energy in the turbine exhaust is utilized to raise the temperature of the compressor discharge flow, thereby reducing the energy input requirement for the combustor (Figure 3-3). The heat transfer is accomplished by means of a regenerator or recuperator. A regenerator is a rotating-disc heat exchanger that alternately exposes the heat exchanger core by sections to turbine exhaust gases and compressor outlet air. A recuperator is a fixed (static) heat exchanger with separate passages for turbine exhaust gases and compressor outlet air.

3.1.2 Historical Background

The fundamental principles of the gas turbine engine were well known by the end of the eighteenth century; however, turbomachinery operating

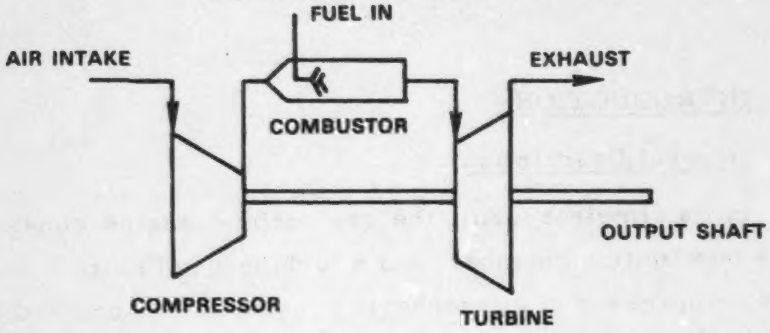
FUEL IN
AIR INTAKE
EXHAUST
COMBUSTOR
OUTPUT SHAFT
COMPRESSOR
TURBINE

Figure 3-1. Simple Gas Turbine Engine, Schematic Diagram

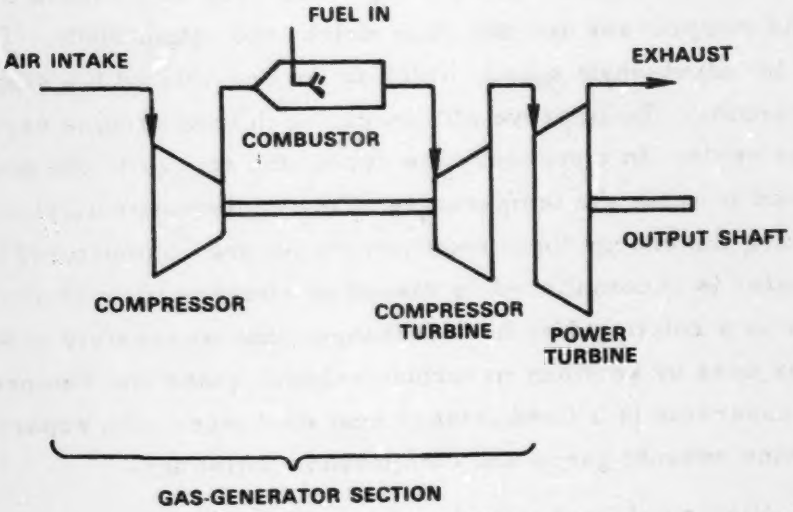
FUEL IN
AIR INTAKE
EXHAUST
COMBUSTOR
OUTPUT SHAFT
COMPRESSOR
COMPRESSOR TURBINE
POWER TURBINE
GAS-GENERATOR SECTION

Figure 3-2. Free-Turbine Engine, Schematic Diagram

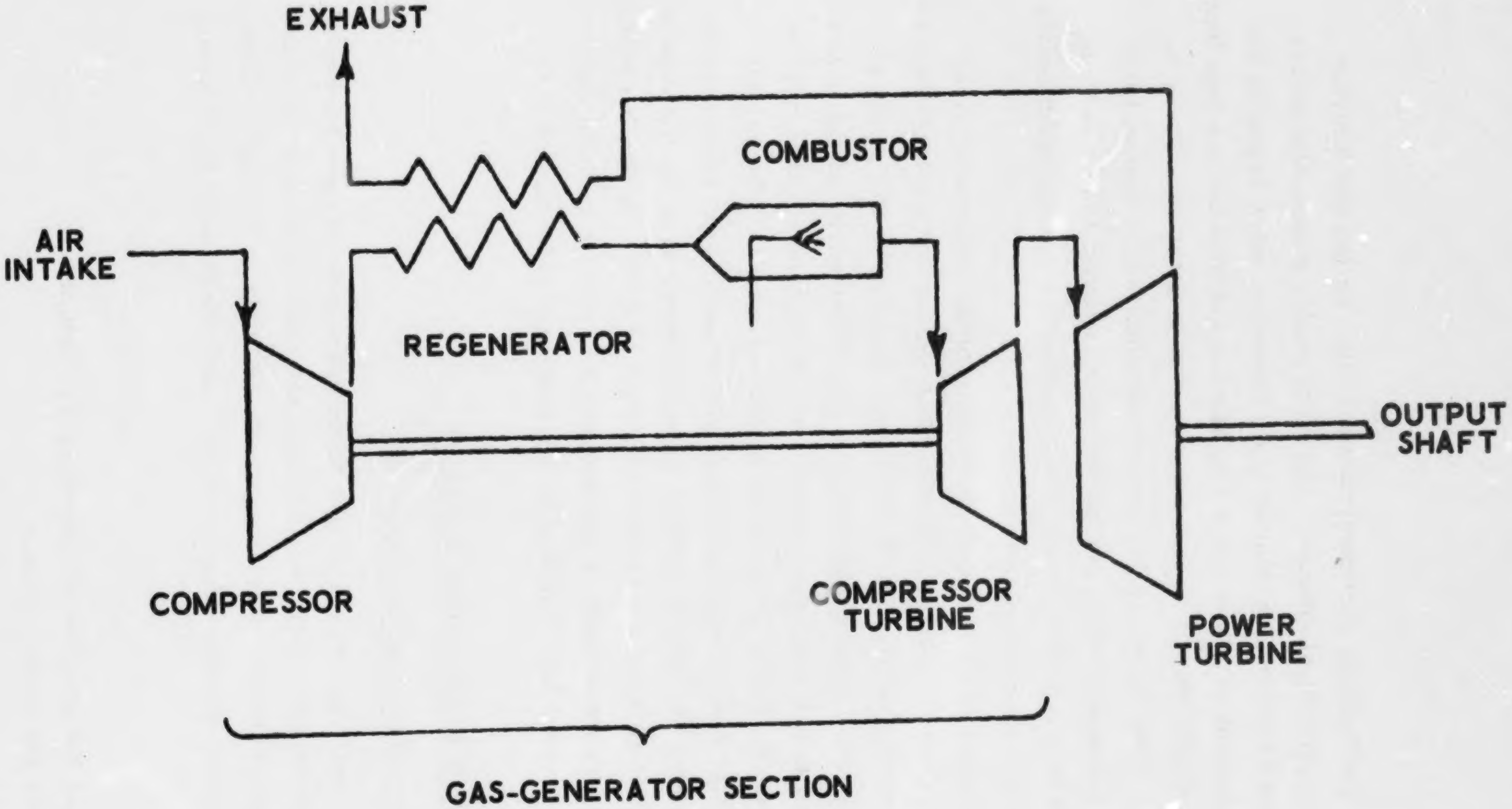
EXHAUST
COMBUSTOR
AIR INTAKE
REGENERATOR
OUTPUT SHAFT
COMPRESSOR
COMPRESSOR TURBINE
POWER TURBINE
GAS-GENERATOR SECTION

Figure 3-3. Regenerative Free-Turbine Engine, Schematic Diagram

successfully on the gas turbine thermodynamic cycle, called the Brayton cycle, is a fairly recent development. The first really successful power plant was the aircraft turbojet developed from intensive work begun in the 1930s. The development of gas turbine engines for automotive use was begun immediately after World War II by Chrysler in this country and Rover in Great Britain. The first Rover turbine-powered automobile ran in 1959. General Motors began automotive gas turbine work around 1948 and Ford about 1950. These developers selected the free turbine configuration early and moved quickly to the regenerative configuration.

Principal early automobile turbine problems were noise, poor fuel economy, poor durability, acceleration lag, and high manufacturing cost. Considerable improvements have been made over the years so that most of these items do not constitute serious deficiencies at the present time.. It is expected that the next generation of gas turbine engines will be further improved, particularly in the area of fuel economy. The recent awareness of domestic energy shortages has placed additional emphasis on improving fuel economy in the gas turbine program sponsored by the Alternative Automotive Power Systems (AAPS) Division of EPA. In the far term, if ceramic turbines are developed, a substantial increase in turbine inlet temperature would be possible, leading to a significant reduction in fuel consumption.

3.2 POWER PLANT DESCRIPTION

3.2.1 Power Plant Configuration

Most existing vehicular gas turbine engines use the so-called free turbine arrangement wherein two turbines are used in series. In this arrangement, a high-pressure, first-stage turbine* drives the air compressor and a low-pressure, second-stage turbine** delivers useful shaft power

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*Also referred to as the gasifier or compressor turbine

**Also referred to as the power turbine

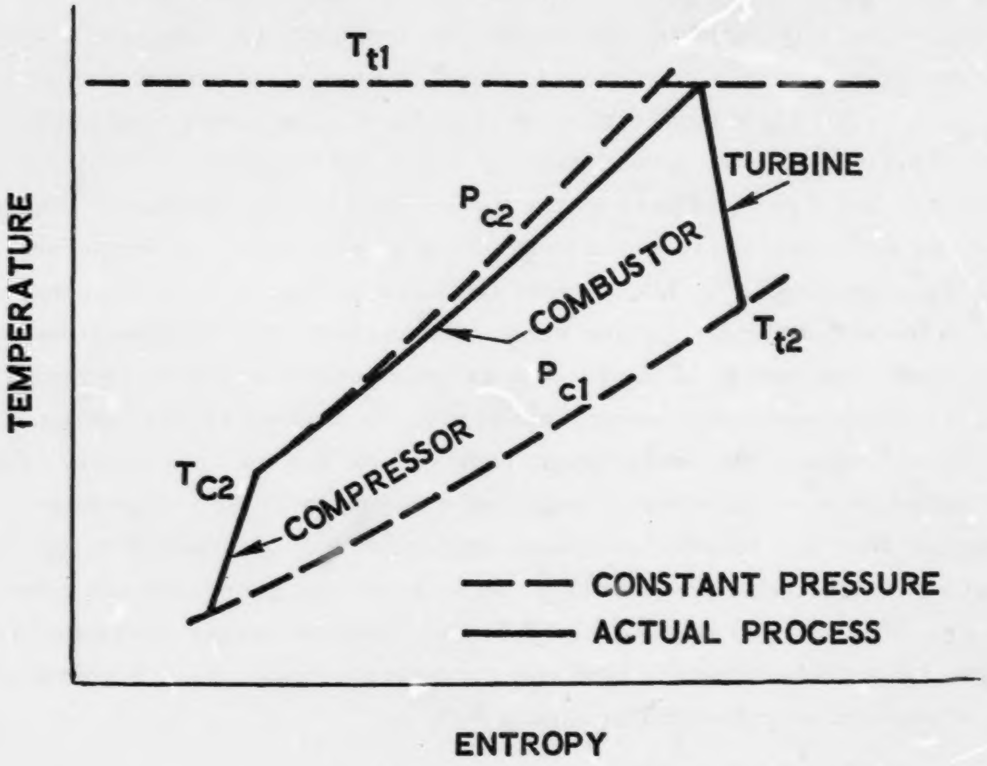
output. This split-shaft scheme allows for flexibility of operation, expecially in conjunction with a variable power turbine nozzle. For example, the variable turbine nozzle may be used to provide engine braking in addition to its value in providing high torque at low turbine speed.

The single-shaft arrangement in which the turbine supplies both the compressor power and the net power output has been used extensively in electric power generation applications where the gas turbine operates at a fixed design speed. The torque/speed characteristics of the single-shaft engine are very steep and the torque is virtually not available below about 50 percent peak speed. Therefore, the configuration requires a wide-range, multiple step or a continuously variable gear-ratio transmission.

A simple gas turbine cycle utilizes a compressor which increases the air pressure, a combustor in which fuel is added and burned with the air, and a turbine powered by the hot combustion products. The turbine drives the compressor and also produces useful work. A temperature-entropy diagram depicting this process is shown in Figure 3-4. The thermal energy in the simple cycle turbine exhaust is wasted. In a regenerative gas turbine cycle, the energy in the turbine exhaust is used to raise the temperature of the compressor-discharge flow which, for a given engine design power level, reduces the energy input requirement for the combustor. The heat transfer is accomplished by means of a regenerator or recuperator. The regenerator is a rotating-disc heat exchanger that alternatively exposes the heat exchanger core by sections to turbine exhaust gases and compressor outlet air. The recuperator is a fixed (static) heat exchanger with separate passages for turbine exhaust gases and compressor outlet air. A schematic of this equipment is presented in Figure 3-5.

3.2.2 Design Features

No automobile turbine engine is in production today, although several experimental gas turbine engines have been developed by automobile and gas turbine engine manufacturers. Some of these engines and their design features are shown in Table 3-1. In addition to these experimental



T_{t1}
TURBINE
P_{c2}
COMBUSTOR
T_{t2}
P_{c1}
T_{C2}
COMPRESSOR
TEMPERATURE
CONSTANT PRESSURE
ACTUAL PROCESS
ENTROPY

Figure 3-4. Temperature-Entropy Diagram for Simple Cycle Gas Turbine

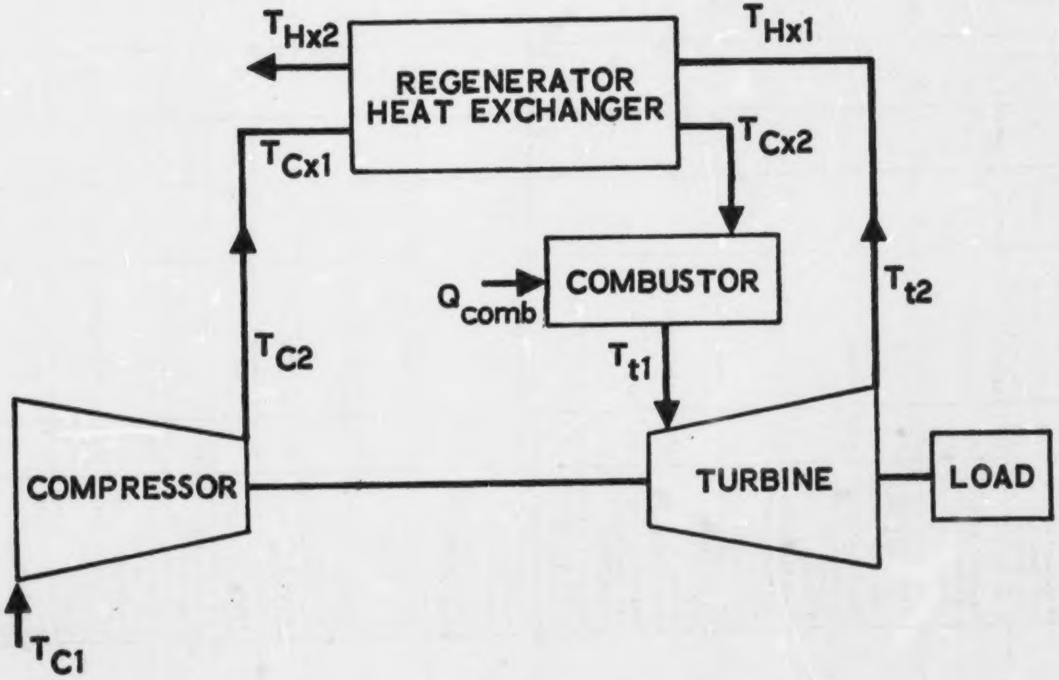

T_{Hx2}
REGENERATOR HEAT EXCHANGER
T_{Hx1}
T_{Cx1}
T_{Cx2}
Q_{comb}
COMBUSTOR
T_{t2}
T_{C2}
T_{t1}
COMPRESSOR
TURBINE
LOAD
T_{C1}

Figure 3-5. Equipment for Gas Turbine Regenerative Cycle, Schematic Diagram

Table 3-1. Engine/Vehicle Basic Characteristics

Manufacturer	Type of Engine	hp	Peak Operating Speed, rpm	Dimensions, in.			Specific Weight lb/hp	Design SFC lb Fuel/ hp Hr
				Length	Width	Height		
Ford	Three Shaft with Intercooling, Reheat, Recuperator		46,500/91,500/ 37,500[2]					
		320	45,300/38,200[3]				5.31	0.41
	Regenerative Cycle with Ceramic Regenerator	450	37,500/31,650	40.8	34.6	41.8	3.78	0.41
		525	37,500/31,650				3.23	0.43
GM	Free Turbine	225	44,000/ ?	?	?	?	?	0.55[4]
	Regenerative Cycle with Metal Regenerator	225	33,000/27,000	37	26.4	24	2.68	0.54
		280	-	36	30	36	3.4	0.45
		325	37,100/ 30,830	47	28	42	5.23	0.475
AiResearch	Single Shaft Regenerative Cycle with Metal Regenerator	155[1]	70,230	-	-	-	2.2	0.41
	Single Shaft Recuperative Cycle with Ceramic Recuperator	125[1]	83,600	-	-	-	3.2	0.43
	Free Turbine Regenerative Cycle with Ceramic Regenerator	175[1]	67,500/55,200	-	-	-	2.5	0.42
United Aircraft	Simple Cycle Single Shaft	150[1]	130,000	17	-	19	1.07	0.51
	Simple Cycle Free Turbine	150[1]	130,000/40,000	21	-	17	1.33	0.51
	Regenerative Cycle with Regenerative Single Shaft	163[1]	112,000	26	-	21	1.78	0.45
	Regenerative Cycle with Regenerative Free Turbine	166[1]	66,500/71,300	27.5	-	28	2.95	0.53
Williams Research	Regenerative Cycle with Ceramic Regenerator	80	61,000/51,000	24	24	16	3.13	0.59
	Regenerative Cycle with Regenerative Free Turbine	130[1]	-	-	-	-	1.4	0.40
Chrysler	Regenerative Cycle with Regenerative Free Turbine	150	44,600/45,700	33	27.5	29.5	4.42	0.60
Rover	Recuperative Cycle with Metal Recuperator	150	65,000/36,000	38	30	27	3.17	0.55
Fiat	Simple Cycle Free Turbine	200	29,000/22,000	-	-	-	2.90	0.95
Volvo	Single Shaft Regenerative Cycle	250	43,000	51.2	26.4	29.5	3.26	0.40
Boeing	Simple Cycle Free Turbine	400	-	39	27	24	1.0	0.75
GE	Regenerative Cycle with Ceramic Regenerator	150[1]	40,000	34.5	26.5	24.1	4.79	0.65
	Recuperative Cycle with Ceramic Recuperator	150[1]	80,000/50,000	31.5	27.0	24.5	4.57	0.50
Solar	Recuperative Cycle with Metal Recuperator Free Turbine	300	36,400/31,000	50	36	41	5.3	0.60

[1] Design Engine, not built or under test

[2] Three speeds shown for low pressure spool, high-pressure spool, and free turbine, respectively

[3] Two speeds shown for dual-shaft (free turbine) engines; first figure for first stage turbine, second figure for power turbine

[4] Experimental test bed engine.

engines, the EPA-sponsored gas turbine studies resulted in several engine designs for a nominal 4,000-pound vehicle. The design features of these engines will be discussed in the following sections.

3.2.2.1 Power

Design power levels for several experimental engines and the EPA-sponsored study engines are shown in Table 3-1. They range from an 80-hp engine for an AMC Hornet designed by Williams Research to a 525-hp truck engine built by Ford. The range of power for the paper-study engines was from 125 to 175 horsepower.

3.2.2.2 Engine Size and Weight

Size and weight of the engine impact the power train packaging, frame, suspension, and other vehicle design elements. In general, both specific weight and specific volume of gas turbine engines are lower than for equivalent spark ignition engines. Weight and volume data for a number of gas turbine engines are provided in Table 3-1. The specific weight, and specific volume values for a family of engines, when plotted against horsepower, may be expected to decrease as the horsepower increases. However, variations in the data shown in Table 3-1 do not permit such a trend to be delineated. Note that the simple-cycle engines are significantly lighter than the regenerative engines.

The dimensional data available do not permit an accurate calculation of engine volume; therefore, only overall engine dimensions are shown in Table 3-1. The simple-cycle engines are also seen to be relatively compact as compared to the regenerative engines. In each category the engine size appears to be inversely related to peak operating speed.

3.2.2.3 Engine Components

The efficiencies of compressor and turbine have direct effect on cycle efficiency and the net work output of the system. An efficient combustor is important from the standpoint of emission and fuel economy considerations. Careful design of regenerators and recuperators in a regenerative cycle is important to ensure high heat exchanger effectiveness and low pressure losses. Other components, such as bearings, have a significant impact on maintaining mechanical integrity of the engine. Engine control components also have to be defined to ensure that proper response and performance designed into the engine are realized over the complete operating map. These components are discussed in the following sections.

3.2.2.3.1 Compressors

The compressor design may be either axial or centrifugal. In large gas turbines of the type used for jet aircraft or stationary power equipment, axial compressors are typically used because of their superior efficiency at the high flow rates of these applications. In an automotive gas turbine, where the maximum airflow rate is approximately 2.5 lb/sec, the blade height of an axial compressor is very small, and maintaining good efficiency is a problem. Hence, in this application, the centrifugal compressor is preferred because, in addition to eliminating the blade-height problem: (1) the centrifugal impeller (wheel) is rugged and easier to manufacture, (2) maintenance problems are fewer, (3) the axial length is shorter (which improves engine packaging and bearing-support loads), and (4) the inertia is lower (which improves engine response).

Important parameters affecting the performance of centrifugal compressors are the clearance between the impeller and the shroud and the ratio of impeller-to-shroud clearance to blade width at the tip of the impeller. In general, the relationship between efficiency loss and tip clearance is linear (Ref. 3-2). Typically, the clearance losses vary between less than 1 percent for large compressors to about 3 percent for very small units.

With regard to impeller materials, aluminum can be used for pressure ratios up to about 4.5:1. Beyond that point, steel alloys and titanium are required in single-stage designs.

3.2.2.3.2 Turbines

Like compressors, either axial or radial turbine configurations may be employed in gas turbines. In large gas turbine engines, the flow rates are high and the turbines used are axial. For automobile gas turbines with maximum flow rates of approximately 2.5 lb/sec, axial or radial turbines are feasible. Radial turbines are more rugged than the axial type, and their performance is less sensitive to variations in tip clearance. They can also be operated at higher speed. In single-shaft arrangements, a radial turbine is preferred. However, for split-shaft designs in which the two turbines are in series, axial turbines are preferred because of packaging considerations for the routing of gas-flow passages. A typical turbine performance map is presented in Figure 3-6 showing the relationships between corrected flow, pressure ratio, and corrected speed.

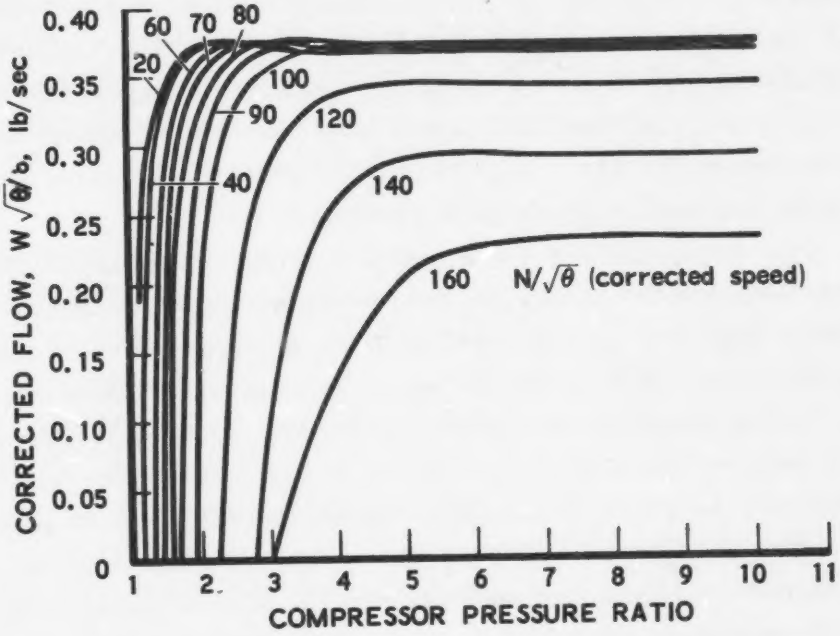


Figure 3-6. Estimated Turbine Performance, Single-Shaft Recuperated Engine (Ref. 3-1)

A critical dimensional tolerance impacting performance as well as mass production quality control requirements is the tip clearance for the axial turbine wheel and the shroud clearance for the radial in-flow turbine.

State-of-the-art turbine design indicates the possibility of nominal design point operation at 1,900°F turbine-inlet temperature for super alloy investment castings without blade cooling. For higher temperatures, blade cooling or advanced ceramic materials would be required.

3.2.2.3.3 Combustors

In the past, when emission control was not a design objective, the prime consideration in gas turbine combustor design was simply to produce the highest overall combustion efficiency in a compact configuration. The basic approach now is to create a hot combustion zone by injecting fuel in a recirculating flow region (primary combustion zone) where near-stoichiometric air/fuel conditions are maintained. The high primary zone gas temperatures are reduced in the secondary zone of the combustor by mixing the combustion products with dilution air.

An example of a conventional combustor configuration is the device selected by Ford for its 707 heavy-duty gas turbine engine. The design, illustrated in Figure 3-7, is a reverse-flow, can-type combustor which utilizes a fuel-atomizing nozzle and an air swirler installed in the dome of the unit. In this design, a strong toroidal vortex is generated along the combustor axis which, according to Ford, assures stable combustion at all operating conditions of the engine. Approximately 20 to 30 percent of the total engine air flow is utilized in the primary zone to provide a primary air/fuel mixture ratio between about 20 and 35. The combustion gases are diluted with secondary air injected downstream of the primary zone in such a manner that a uniform flow velocity profile is obtained at the entrance to the gasifier. This design satisfied Ford's initial objectives for stable combustion, high combustion efficiency, low pressure drop, long life, low cost, and low emissions.

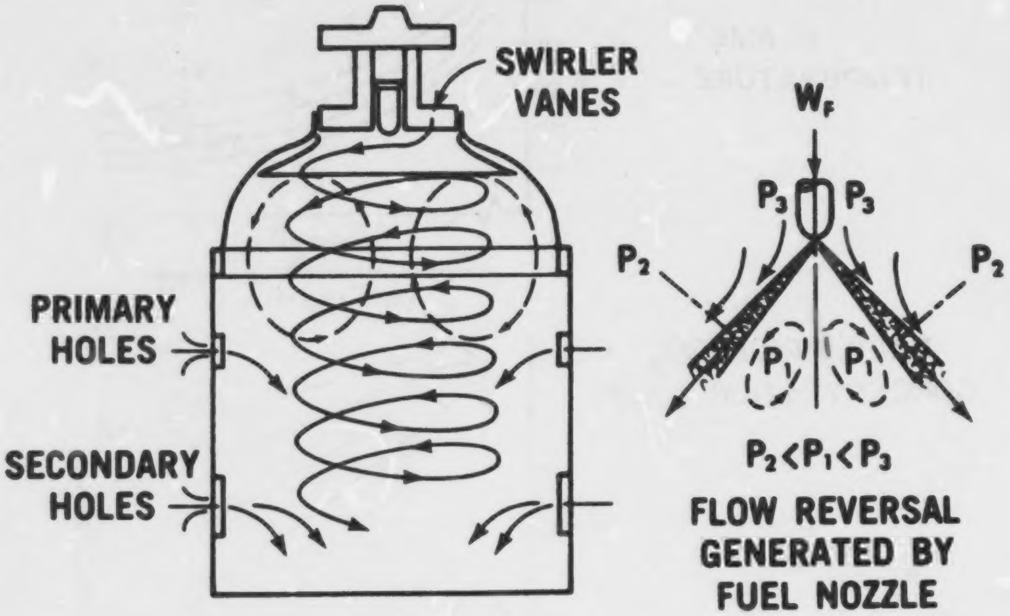

SWIRLER VANES
PRIMARY HOLES
SECONDARY HOLES
W_F
P_3
P_3
P_2
P_2
P_1
P_1
P_2<P_1<P_3
FLOW REVERSAL GENERATED BY FUEL NOZZLE

Figure 3-7. Flow Path for Reverse-Flow, Can-Type Combustors (Ref. 3-3)

Present goals in automotive gas turbine combustor design are to meet or better the original 1976 Federal emission standards while maintaining good fuel economy. A major problem arises from the fact that superior fuel economy requires regenerative heating and/or high compressor pressure ratio, resulting in high combustor temperatures conducive to high NO_x emissions. Figure 3-8 illustrates this relationship. As the combustor-inlet temperature is increased (for the same equivalence ratio) the combustor temperature and the NO_x levels increase. To minimize the formation of NO_x, the following design principles are currently being evaluated by industry (Refs. 3-4 through 3-7):

a. Lean primary zone operation

b. Prevaporization of the fuel before combustion

c. Air/fuel ratio control by means of variable geometry air ports.

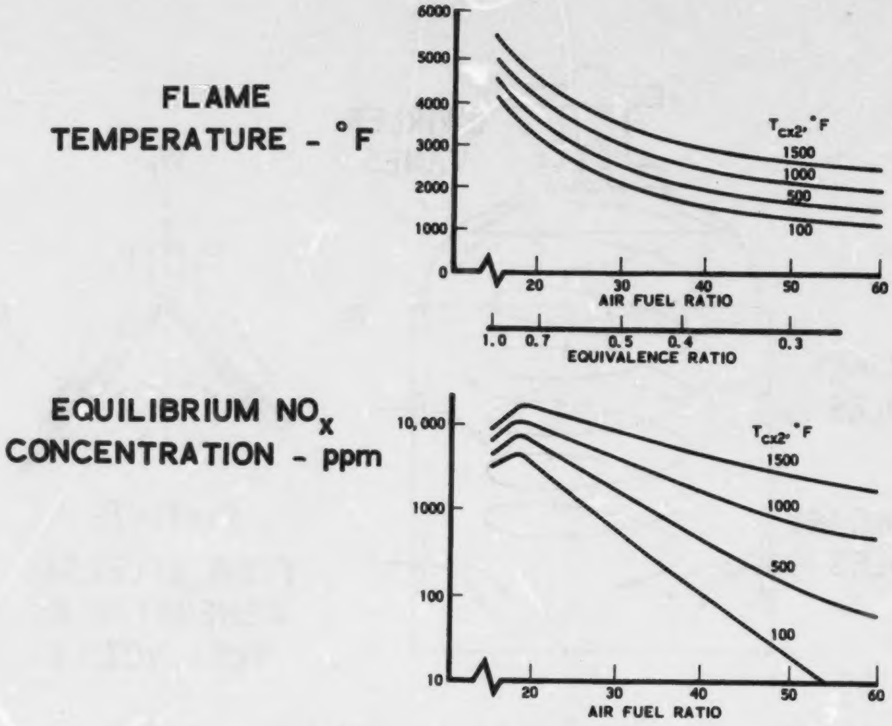


Figure 3-8. Estimated Flame Temperatures and Equilibrium NO_x Concentrations as Functions of Air Fuel Ratio for Various Combustor Inlet Temperatures (Ref. 3-4)

An example of a fixed-geometry combustor configuration is the current Chrysler combustor. The primary zone reaction volume is very small, in order to minimize the residence time of the combustion gases at the high combustion temperatures and, hence, to minimize the formation of NO_x. Secondary air is injected immediately downstream of the primary zone to quench the NO_x reactions and complete the hydrocarbon (HC) and carbon monoxide (CO) oxidation reactions.

3.2.2.3.4 Regenerators and Recuperators

Regenerators are rotating heat exchangers which operate by alternately exposing the core sections to the "cold" and "hot" compressor and turbine exhaust flows. As illustrated in Figure 3-9, a barrier seal

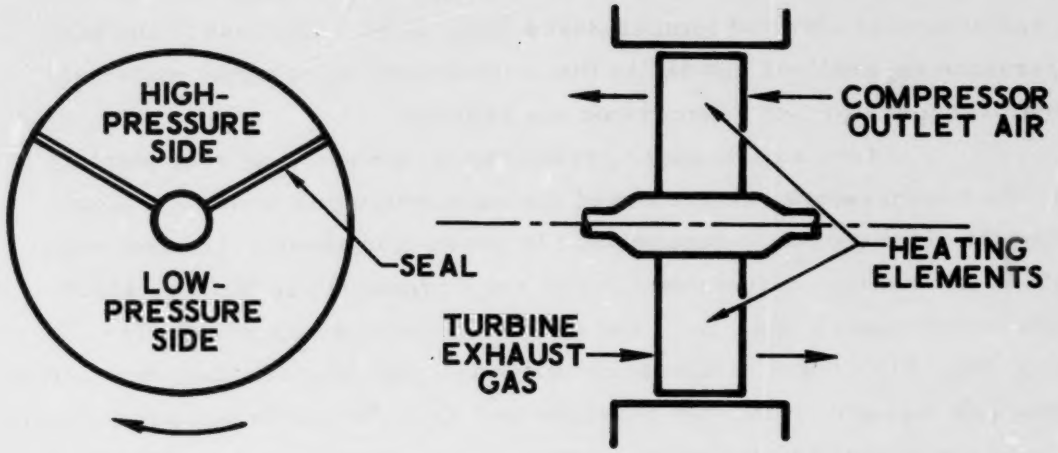


Figure 3-9. Disc-Type Rotary Regenerator (Ref. 3-1)

separates the "cold" and "hot" sides of the heat exchangers in which both metallic and ceramic cores have been used. The effectiveness of these rotating heat exchangers, defined as the ratio of heat transferred into the cold fluid to the amount of heat available in the hot fluid, can be very high. Values of the order of 0.9 have been achieved with regenerators designed for automotive gas turbine applications.

In the past, leakage from the high-pressure side to the low-pressure side has been a major problem with regenerators; however, this now appears to be solvable.

Recuperators are fixed-boundary, stationary heat exchangers. For automotive applications, the effectiveness of these heat exchangers can be as high as 0.85. Compared to regenerators, recuperators may be bulky but require no rubbing seals or rotating drive. Materials used in the manufacture of recuperators include mild steel, stainless steel, Inconel, and Hastelloy (Ref. 3-8). Mild steel has been used for low-temperature applications in stationary gas turbine plants. An excellent recuperator material is 347 stainless steel, which has good creep strength and hot erosion resistance at elevated temperatures (Ref. 3-8). The cost of the high-temperature superalloys has led to the investigation of ceramic materials for both recuperator and regenerator applications.

From a packaging, performance, weight, and cost point of view, the regenerators are preferred for automotive gas turbines. Currently, the glass-ceramic regenerator is growing in favor. The two major suppliers of ceramic regenerator cores are Corning Glass Works, which manufactures Cercor (Ref. 3-9), and Owens-Illinois, which makes Cer-Vit (Ref. 3-10). Both manufacturers utilize honeycomb construction; the Cercor product has straight triangular passages and Cer-Vit has hexagonal passages. The advantages claimed for glass-ceramic regenerators are compactness, low density, and low cost. The Cercor density runs between 30 to 42 lb/ft^3. The cost of a 17-inch diameter Cercor regenerator is projected to be about $40 based on an annual production rate of 100,000 units.

The employment of such ceramic materials requires an understanding of their properties and limitations as well as the development of satisfactory component design techniques. A major problem area involves thermal stresses. Establishment of material and design requirements necessitates a careful heat-transfer analysis of steady running and transient operation to determine thermal stresses. For a regenerator application, where the very nature of its function demands a high-temperature gradient, glass ceramics of the lithium-alumina-silicate (LAS) type with their very low thermal expansion are very attractive.

Another consideration in material selection is resistance to chemical attack. Oxidation is the most obvious concern, but combined salt ingestion in combination with small amounts of sulfur in the fuel creates at high temperature a corrosion condition known as sulfidation. LAS is susceptible to such corrosion but intensive development is under way to provide satisfactory protective coatings or substitute materials.

Chrysler experimented with ceramic regenerators as early as 1950, when it dropped the ceramic approach in a decision to use all-metal regenerators in its fourth- and sixth-generation automobile gas turbine engines. Chrysler feels that metal regenerators have lower seal friction and are less expensive at current production levels than ceramic configurations. In terms of heat loss, ceramic regenerators are considered superior. Even though Chrysler continues to favor metallic regenerators for reasons of mechanical performance, reliability, and cost, they are nevertheless working on the development of improved ceramic designs.

3.2.2.3.5 Bearings

Both rolling element and journal bearings are used in gas turbine engines. The rolling element bearing is preferred in aircraft gas turbines because of superior friction characteristics under variable temperature and load conditions. In stationary gas turbines, journal bearings are used to take advantage of their long service life (Ref. 3-11). For automobile gas turbines, both journal and rolling element bearings are being considered.

3.2.2.4 Materials

The production potential of gas turbine engines is dependent on the types, cost, and availability of materials as received at the engine plant and on the processes used to manufacture the engine. These factors are reflected in production rates, equipment design, facilities, and the eventual production cost of the engine. The following paragraphs discuss the availability of raw materials and the requirements of processed materials as well as their impact on supply industries, engine manufacturing processes, equipment tooling and facility requirements, engine production costs, and possible means for reducing production costs. Most of these items are compared to current and future emission-controlled piston engines so as to provide a basis for assessing effects incurred by a changeover to gas turbine engine manufacture.

The piston engine without advanced emission controls has temperature/stress requirements that can be readily satisfied by the use of low-alloy carbon steels, cast iron, and aluminum; piston engines with such emission controls, as presently conceived, will require a certain amount of high-temperature metal in the exhaust system. In contrast, the gas turbine engine operates continuously at high temperatures and stresses and, therefore, requires a significant amount of materials having high-strength properties at elevated temperatures. These considerations govern the difference in basic materials selected for the different engines.

Table 3-2 provides an approximate comparison of the types and weights of materials used for contemporary and future piston engines as well as for a low-pressure-ratio regenerative gas turbine; this comparison is based on data provided in Reference 3-12. Data regarding the use of Inconel 718 and 738 have been deleted from Table 3-2 because, by comparison with Inconel 713, Inconel 718 contains much greater levels of niobium, and Inconel 738 contains much greater levels of tungsten and tantalum. Most manufacturers have elected not to use these potentially critical elements for their engine designs. It should be noted also that there are considerable variations in material weights for gas turbines proposed by a number of manufacturers. However, compared to piston engines, all concepts use relatively large amounts of high-alloy steel, Inconel, and Hastelloy.

Table 3-3 compares the 1969 U.S. consumption of potentially critical materials with the material needs for production of 10 million engines (the approximate quantity of passenger automobile engines produced annually in the United States in recent years). These data were extracted from various sections in References 3-1, and 3-12 through 3-15, and are based on initial quantities of materials required with fabrication-scrap recycling. As can be seen, some of the material requirements for the gas turbine engine, when contrasted with 1969 U.S. consumption levels, would appear to present a significant impact on future availability of these materials. However, some lessening of the impact is anticipated. First, inquiries to stainless steel producers have revealed that there has been a significant increase in

Table 3-2. Engine Material Content and Weight (based on Ref. 3-12)

Material	lb/engine (scrap weight not included)		
	Piston Engine		Gas Turbine
	Circa 1970	Circa 1976 (Est.)	4:1 Regenerative (Est.)
Gray Cast Iron	220	224	--
High-Temp Cast Iron	--	28	23
Doctile Cast Iron	20	20	252
High Alloy Steel	5	5	65
Low Alloy and Carbon Steel	189	226	68
Copper	19	19	15
Brass	25	25	--
Stainless Steel	--	5	14
ACI Type HU Steel	--	25	--
Aluminum	20	21	4
Inconel 713	--	--	8
Hastelloy-X	--	--	4
Cercor (Al_2O_3)	--	--	20
Platinum/Palladium/ Ruthenium	--	0.007	--
Total	498	598	474

Table 3-3. Estimated High-Temperature Metal Element Requirements for 10 Million Engines* (Refs. 3-1, 3-12 through 3-15)

(Millions of Pounds)

Element	Annual Needs			Usage and Availability		
	Piston Engine		Gas Turbine	United States		World
				1969	1972	1970
	Circa 1970	Circa 1976	4:1 Regenerative	Annual Consumption	Stockpile	Reserves
Chromium	8	147	54	481	5331	1,550,000
Nickel		141	124	283	79	150,000
Cobalt			1.4	15	74	4,818
Molybdenum			8.5	52	43	11,000
Tungsten			2.4	16	129	2,800
Niobium			1.6	3	9	13,000
Platinum Type		0.1		0.1	0.14	31

*Amounts contained in the engine after fabrication availability of chromium and nickel since 1969. Second, a further expansion in production is expected in order to meet the requirements for the 1976 piston engine, unless some drastic changes in the present emission control concepts occur. Therefore, cobalt, niobium, and molybdenum are the only materials that may be critical in availability.

The above analysis for Table 3-3 considered the initial demand for critical elements but did not include the recycling of scrapped engines to extract the critical elements. It is estimated that, for the current annual engine production rates, the quantities of high-temperature material elements to be obtained from ore could be eventually reduced by a factor of 2 to 3 by recycling all of the scrapped engines. Since the introduction of gas turbines would probably build up from initial production quantities of approximately 250,000 engines per year to the full potential in excess of 10 million, the material requirements in Table 3-3 eventually would not have to be obtained from ore alone were recycling to prove profitable.

Aside from problems of availability, consideration must be given to (1) potential effects on cost of the high-temperature elements due to supply/demand factors, and (2) the balance-of-trade effect on the U.S. economy due to the requirements of increased purchases of ore or refined metals from foreign countries.

3.2.2.5. Fuel Requirements

A gas turbine can use either liquid or gaseous fuel. Liquid fuel requires atomization for efficient burning; the gaseous fuel does not require any special preparation. Gas turbine engines can use gasoline, kerosene, or diesel fuel. The same burner in the gas turbine engine can accept several types of liquid fuel, possibly without adjustments for a family of hydrocarbons. In using gaseous fuel, a different type of orifice system is required.

3.2.2.6 Pressure Ratio and Temperature

To increase the net work output of the gas turbine, the compressor work has to be reduced, the turbine work has to be increased, or both. An increase in the turbine inlet temperature has a significant effect on the turbine work output. However, the operational turbine inlet temperature is limited by material properties. Present-day gas turbines can operate at turbine-inlet temperatures of 1,700°F continuously and at temperatures as high as 1,900°F for a short time. As will be discussed later, with ceramic turbine materials this temperature can be increased at 2,500°F.

As the engine-pressure ratio is increased at constant turbine inlet temperature, the temperature difference between the turbine and compressor discharge decreases and regenerative heating becomes less significant. For this reason, the design point efficiency of the simple cycle optimizes at high pressure ratio, and the efficiency of the regenerative cycle optimizes at low pressure ratio. This performance effect is shown in Figures 3-10 and 3-11, where the brake specific fuel consumption (BSFC) is plotted as a function of compressor-pressure ratio for a range of turbine inlet temperatures for typical simple and regenerative cycles, respectively. However, the selection of pressure ratio in design studies will be influenced by engine performance at part-load operation--where a major portion of operating times is expected to be experienced by engines powering automobiles to be driven in urban traffic conditions.

In all cycles, the pressure ratio available in the turbine is reduced by the combustor-pressure drop. In the regenerative cycle, additional pressure losses occur in the "cold" and "hot" flow passages of the heat exchanger, which further reduces the turbine-pressure ratio and the turbine work output.

3.2.2.7 Transmission Systems Design Approach

The gas turbine engine, as described in previous sections, generally operates at speeds above 30,000 rpm, while the maximum engine output speed required for an automobile is about 5,000 rpm. Hence, the turbine shaft speed has to be geared down to a level practical for automobile drive. This is accomplished by a fixed-ratio, reduction gear box installed ahead of the transmission.

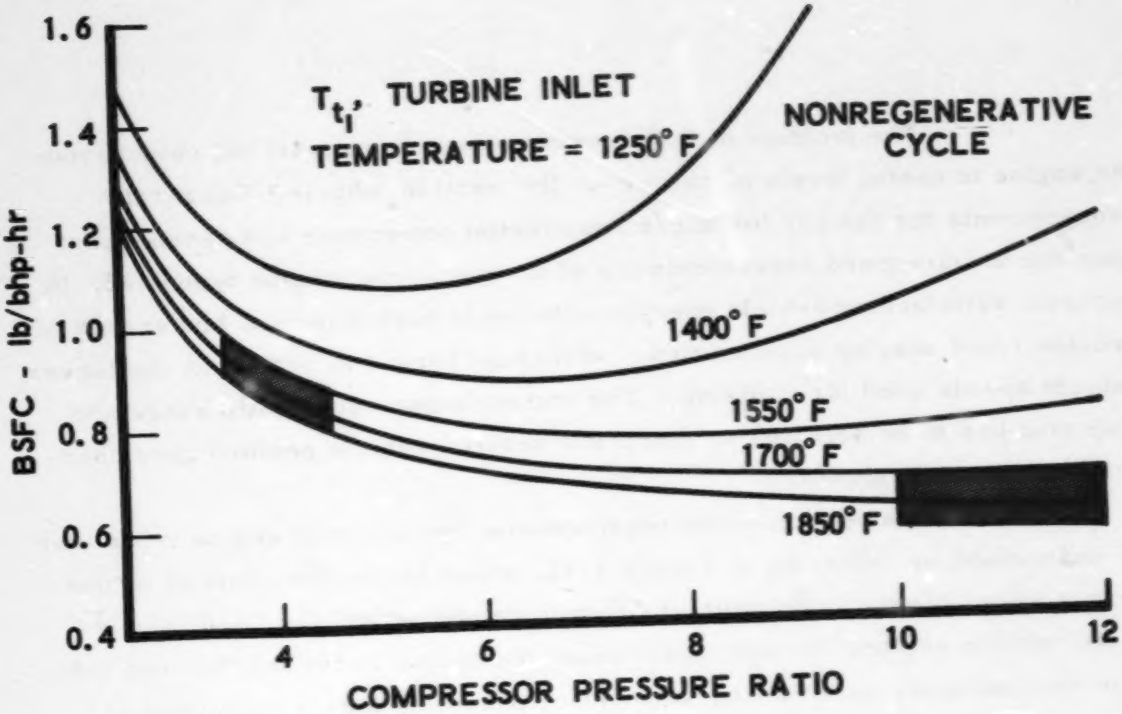


Figure 3-10. Estimated Effect of Pressure Ratio and Turbine Inlet Temperature on Fuel Economy for Nonregenerative Cycle (Ref. 3-16)

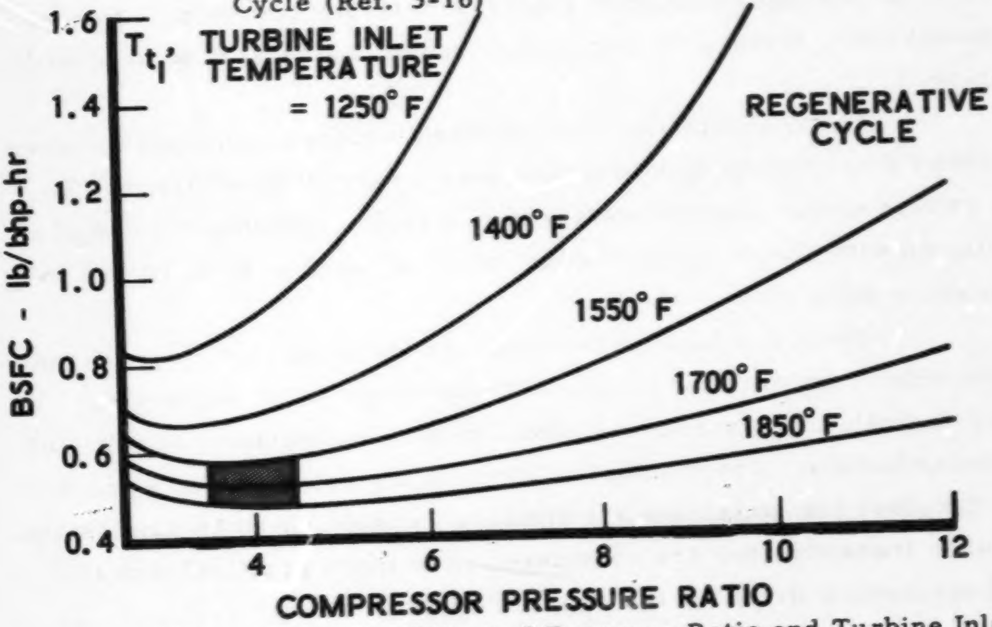


Figure 3-11. Estimated Effect of Pressure Ratio and Turbine Inlet Temperature on Fuel Economy for Regenerative Cycle (Ref. 3-16)

The function of the transmission is to convert the output from the engine to useful levels of torque at the vehicle wheels. Gear-ratio requirements for the gas turbine/transmission powertrain are dependent upon the torque-speed characteristics of the particular engine type used. In general, satisfactory vehicle operation demands high torque at low speeds to provide rapid startup acceleration, while less torque is needed at the higher vehicle speeds used for cruising. The transmission speed ratio range and step size has to be selected so that these desired vehicle performance characteristics are achieved.

The transmission requirements for different engine types may be understood by referring to Figure 3-12, which shows the trend of torque versus speed for the free (split-shaft) turbine engine and the fixed (single-shaft) turbine engine. It may be seen that the torque curve for the free turbine approximates an ideal characteristic, providing maximum torque at low speed while maintaining a relatively high level of torque over the speed range. The general characteristics of this curve are similar to the characteristics of an internal combustion engine with torque converter. Hence, conventional two-, three-, or four-speed transmissions may be used with this engine.

The characteristic for the fixed turbine engine, on the other hand, shows a very steep rise in torque over a narrow speed range. To satisfy vehicle torque requirements with this engine demands the use of a transmission with a wide range in gear ratios; a range of up to 10 to 1 in eight or more steps may be required.

There are four major types of wide speed-ratio range transmissions under consideration for use with the single-shaft engine: the hydromechanical, the 8-speed mechanical drive, the traction, and the belt drive transmissions. The mechanical drive is a stepped variable speed device; the other transmissions are stepless variable speed transmissions. These other transmissions are considered to be more practical than the 8-speed mechanical drive for the single-shaft engine.

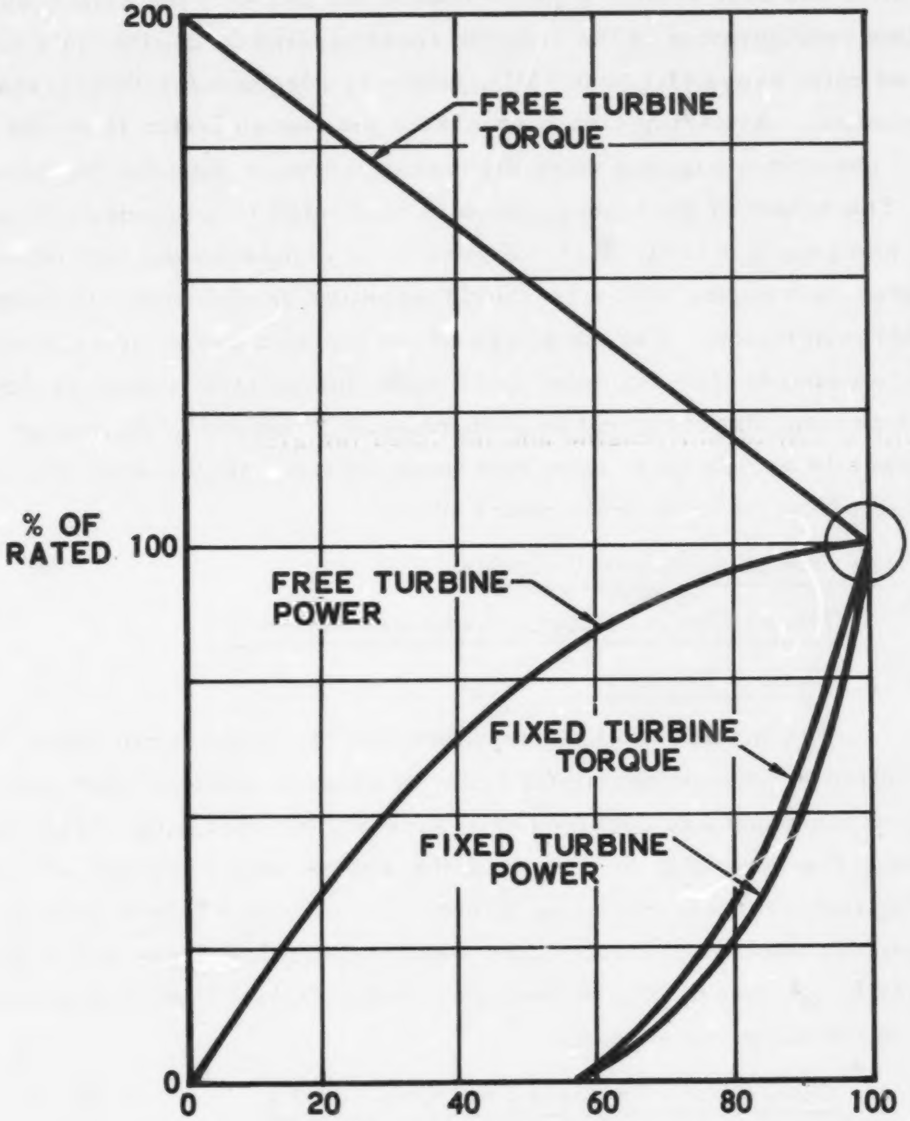


Figure 3-12. Typical Fixed and Free Turbine Torque and Power Characteristics

Considering the state of the art, the traction transmission seems to offer the best available performance for use with the single-shaft engine. One configuration of the traction transmission is capable of a total of 9:1 speed ratio range (3:1 to 0.33:1), which is adequate for single-shaft engine operation. A startup clutch separates the transmission from the gear box. The clutch engages when the transmission is put into the drive position. The output of the transmission is connected to a standard torque converter and gear box (Ref. 3-1). Figure 3-13 compares the fuel economy of the single-shaft engine with a traction transmission and with a four-speed automatic transmission. The advantage of the traction device is especially evident at low speeds; for example, at 10 mph, the traction transmission shows a 25 percent improvement in performance. However, the design feasibility has to be proven in extensive tests before a single-shaft engine can be selected for an automobile power plant.

3.2.3. Operating Characteristics

3.2.3.1 Power Control Requirements and Methods

3.2.3.1.1 Control Devices

The turbine inlet temperature and the engine shaft speed have to be controlled to prevent engine failure. In addition to these limit controls, other control functions are required to ensure proper operation of the engine at all times. For example, at part load the engine would operate at reduced turbine inlet temperature, resulting in poor fuel economy unless certain control devices, such as variable compressor-inlet-guide vanes and variable turbine nozzles are used. These devices and the overall control system are discussed in the following sections.

3.2.3.1.2 Compressor Variable Inlet Guide Vanes

The compressor performance is governed by three basic parameters: rotational speed, pressure ratio, and flow rate. At low flow rates, compressor surge effects can produce a significant reduction in compressor

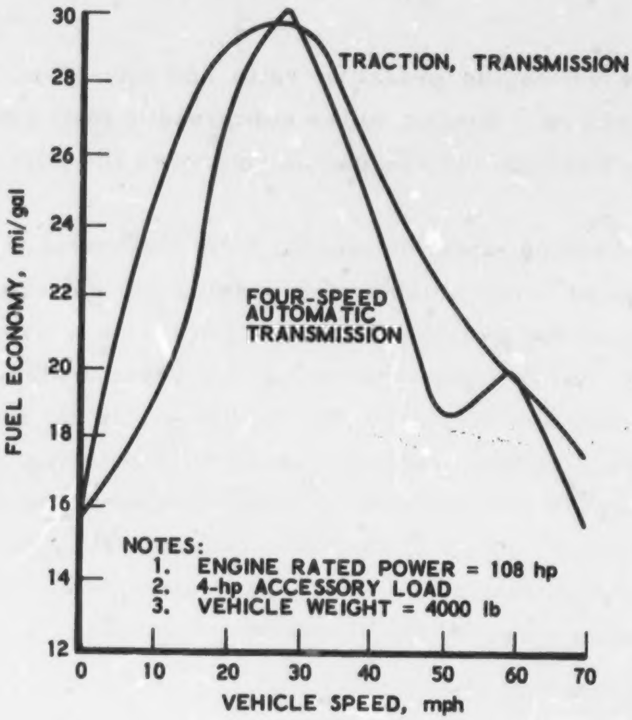


Figure 3-13. Estimated Road-Load Fuel Economy of Single-Shaft Regenerated Engine, 85°F Sea-Level Day (Ref. 3-1)

efficiency. Incorporation of variable inlet guide vanes alleviates these problems to some degree. By adjusting the position of these guide vanes, the air flow of the compressor can be reduced without affecting the rotational speed and pressure ratio of the engine. The turbine-inlet temperature can then be maintained at or near the design point value, and this results in marked improvements in specific fuel consumption at part load.

3.2.3.1.3 Turbine Variable Nozzle

The use of a variable geometry nozzle in the power turbine of the dual-shaft system permits changes to be made in the power split between the power turbine and the first-stage turbine. This allows flexibility in control of the overall engine pressure ratio and operation at optimum turbine inlet temperature. Similar to the compressor inlet guide vanes, incorporation of the variable turbine nozzle improves the part load fuel economy.

The braking effect obtainable from the variable nozzle feature is illustrated in Figure 3-14, which shows braking force characteristics as a function of car speed for two of Chrysler's free turbine, variable nozzle engine designs (Ref. 3-17). The most recent Chrysler design, the sixth-generation engine, was modified from the fourth-generation design to achieve better braking characteristics. Also shown in the figure is the braking force of a V-8 engine-powered vehicle with the transmission in the drive position. As indicated, the braking power of the sixth-generation gas turbine engine is substantially higher than that of the fourth-generation engine and slightly better than the braking power of the standard automobile.

3.2.3.1.4 System Controls

System controls involve the overall regulation of component interactions during engine response to driver commands so that all critical operating variables are maintained within safe limits. The major elements of the control system could include the following items (Ref. 3-1):

- Fuel Control
- Turbine Inlet Guide Vane Actuators
- Compressor Inlet Guide Vane Actuators
- Automatic Starting and Limit Protection Control
- Transmission Control
- Hydraulic Clutch Control
- Power Boost Control
- Sensors.

Control operation could be hydromechanical or electronic. The major difference between the two systems is in sensors, metering sections, and controller elements. In general, hydromechanical control systems use a rotating shaft

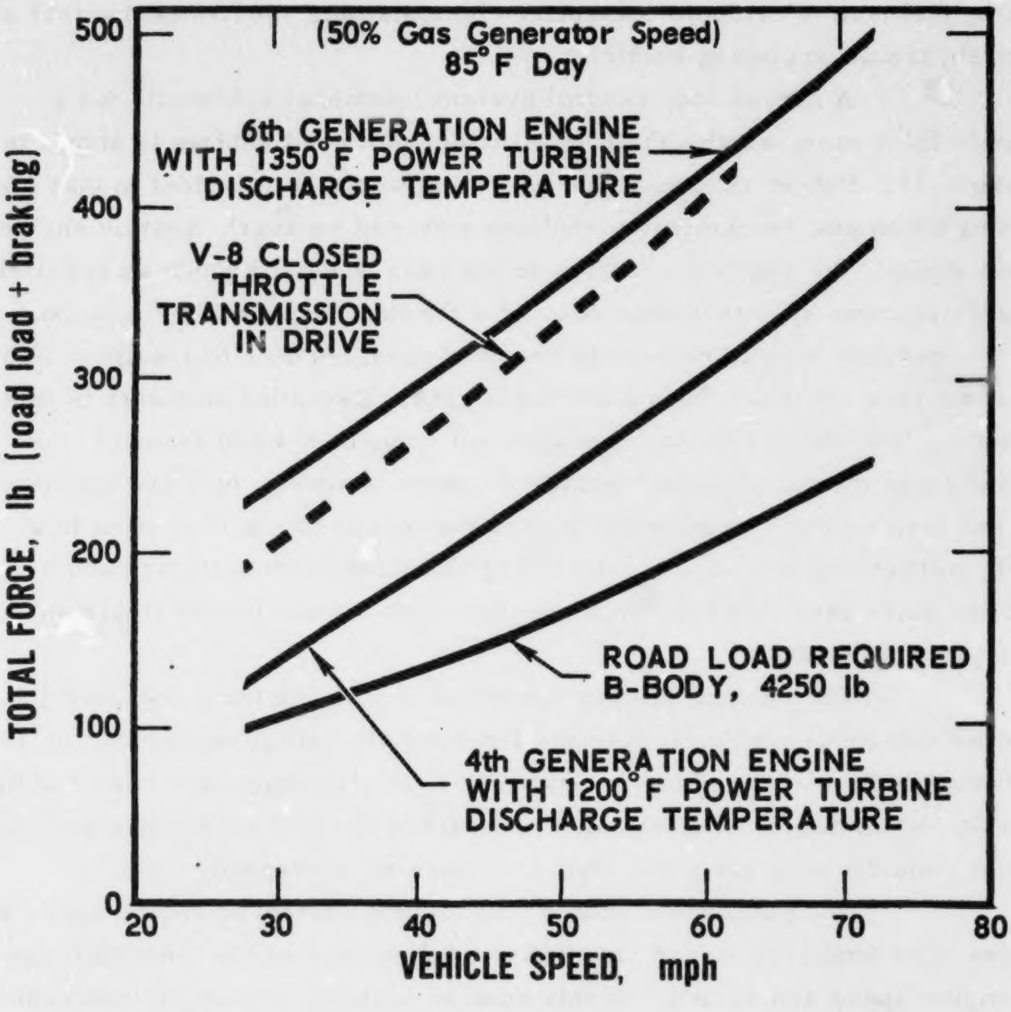


Figure 3-14. Engine Braking Force Characteristics (Ref. 3-17)

and flyweights as the speed-sensing element and a controller to position a metering valve. Electronic control systems have an electromagnetic speed-sensing element, electronic controller elements, and electromechanical or electrohydraulic metering sections.

A closed-loop control system functional schematic for a variable inlet vane, single-shaft, gas turbine-powered vehicle is shown in Figure 3-15. Driver command and operation would be identical to that for existing automatic transmission vehicles powered by spark ignition engines. In this design, the engine is coupled to the rear wheels through an infinitely variable traction-type transmission and a forward/reverse-type gearbox. Engine operation with zero vehicle speed is provided by a hydraulic clutch that decouples the drive train from the engine. Essential elements in this system include the engine fuel management components and transmission control components. Although somewhat more complex, fuel management of a gas turbine can be compared in function to a carburetor system in a spark ignition engine. Other controlling elements include items such as the inlet guide vane control and the power boost control (water injection) for maximum acceleration.

The control system functional schematic for a free-turbine powered vehicle is comparable to the single-shaft configuration; the major difference is in the transmission control. This difference is a result of the three-speed automatic transmission considered for this application and the special requirements for preventing free-turbine overspeed.

Ford utilizes a double closed-loop control of engine speed and turbine inlet temperature for automotive applications where both shaft power and engine speed are varied. In this control system, predetermined schedules of first-stage turbine speed versus power level and turbine inlet temperature are built into the control system. The system senses first-stage turbine speed and turbine inlet temperature and regulates the power turbine nozzle angle to match the prescribed temperature. Optimum part-load fuel

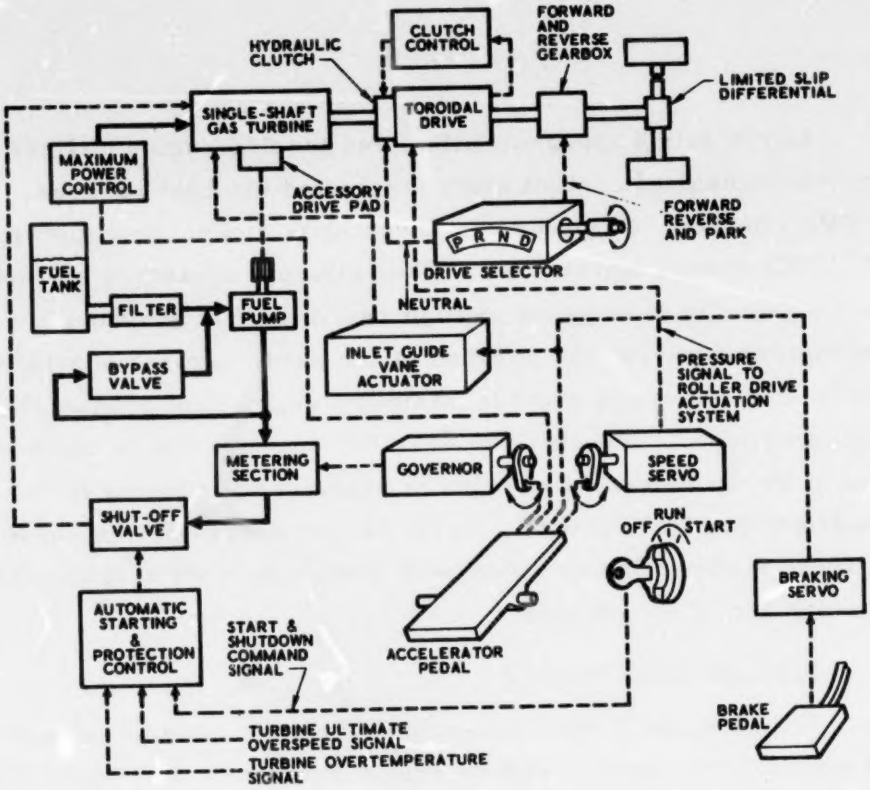

CLUTCH CONTROL
FORWARD AND REVERSE GEARBOX
HYDRAULIC CLUTCH
LIMITED SLIP DIFFERENTIAL
SINGLE-SHAFT GAS TURBINE
TOROIDAL DRIVE
MAXIMUM POWER CONTROL
ACCESSORY DRIVE PAD
FORWARD REVERSE AND PARK
P R N D
DRIVE SELECTOR
FUEL TANK
FILTER
FUEL PUMP
NEUTRAL
INLET GUIDE VANE ACTUATOR
BYPASS VALVE
PRESSURE SIGNAL TO ROLLER DRIVE ACTUATION SYSTEM
METERING SECTION
GOVERNOR
SPEED SERVO
SHUT-OFF VALVE
RUN
OFF
START
BRAKING SERVO
AUTOMATIC STARTING & PROTECTION CONTROL
START & SHUTDOWN COMMAND SIGNAL
ACCELERATOR PEDAL
BRAKE PEDAL
TURBINE ULTIMATE OVERSPEED SIGNAL
TURBINE OVERTEMPERATURE SIGNAL

Figure 3-15. Control System Functional Schematic, Single-Shaft Gas Turbine Engine (Ref. 3-1)

economy is maintained by operating at the highest turbine inlet temperature consistent with the required compressor-surge margin. By adjusting the nozzle vane position, the design-point, turbine-inlet temperature can be maintained for engine speeds as low as 85 percent of the design value. Below that point, the turbine inlet temperature is reduced to avoid compressor surge (an unstable flow condition).

In case of overtemperature, the fuel supply to the engine is reduced. Overspeed protection is provided by reducing the fuel flow and by adjusting the nozzle vanes to the braking position.

Chrysler's sixth-generation free turbine engine utilizes an open-loop hydromechanical control system selected for cost reasons. Under its current EPA contract, Chrysler will investigate closed-loop electronic fuel control (EFC) system approaches. The principal advantage of closed-loop EFC is its capacity to improve system response. In addition, the turbine inlet temperature and the position of the power turbine nozzle can be optimized in this system to provide minimum engine fuel consumption at all operating conditions. Chrysler prefers EFC systems but is concerned about the life of the temperature sensors (a problem not present in the open-loop hydromechanical system). In its 50-car program, the sensors failed after about 5,000 miles as a result of oxidation of the thermocouple beads and/or erosion of the shields.

3.2.3.2 Starting Requirements

Unlike a spark ignition engine, the gas turbine engine requires a design for automatic control to assure engine protection during startup. Engine protection is necessary because certain system malfunctions could result in overtemperature or overspeed. These two conditions are not significantly critical in a spark ignition engine, because combustion takes place with the air fuel ratio less than stoichiometric. Thus, an increase in fuel (failed carburetor) results in a decrease in combustor temperature. However, the opposite effect is true of the gas turbine engine. Also, overspeed (a runaway condition) is less likely in a spark ignition engine, since the combustion chamber and valve train tend to limit the rotational speed of the engine.

3.2.4 Packaging and Installation

Generally, the specific volume (ft^3/hp) of gas turbine engines tends to be lower than that of spark ignition reciprocating engines. Therefore, there should be no problem in fitting the engine package into a conventional-sized engine compartment unless combustor or regenerator volumes were to increase substantially. The engine could be located either in the front or in the rear of the vehicle. A front installation with front-wheel drive or a rear installation with rear-wheel drive would provide improved road traction and would eliminate the long driveline required in conventional car designs. A rear installation would also eliminate the long exhaust ducts and the resultant chassis redesign required to accommodate the duct volume. Although the engine itself is compact, the air-intake and exhaust ducts are large (the gas turbine has five times the air flow of an internal combustion engine), and these have to be packaged properly to meet reasonable envelope requirements and to provide sufficient ground clearance. Ford indicated that packaging the turbine-exhaust system was a problem. Installation of gas turbine engines in conventional vehicles required tearing up of the car-floor pans. (Were the engine less sensitive to exhaust back-pressure, smaller ducts could be used.) Williams Research indicated that the exhaust system for the turbine car could cost twice as much as for the conventional car.

3.3 PERFORMANCE CHARACTERISTICS

3.3.1 Exhaust Emissions

Both Chrysler and Williams Research have tested their gas turbine automobiles for emissions over the Federal Driving Cycle (FDC). Results of these tests are presented in Table 3-4. In these tests, the engines were equipped with conventional state-of-the-art combustors. The Chrysler results meet the CO and HC standards, but not the NO_x standard. The two Williams Research gas turbine engines (WR-25 and 131-Q) installed in a 1971 Hornet and a 1965 Volkswagen showed a high level of exhaust emissions. These data were extracted from References 3-19 and 3-20. The computed performance of other conventional combustors operated over a simulated FDC are also listed in Table 3-4; none of these meet the 1976 NO_x requirement.

A number of EPA contractors have been working on advanced-design combustors. Based on experimental component testing, the emission levels for engines equipped with these devices are predicted as shown in

Table 3-4. Exhaust Emission Data over the Federal Driving Cycle

Engine/Manufacturer	Emissions (gm/mi)			Remarks
	CO	HC	NO_x	
a. Conventional Combustors				
United Aircraft Research Laboratories RGSS-6	0.53	0.15	2.72	Ref. 3-21 Calculated values over simulated Federal Driving Cycle, based on emission data from GM engine GT-309.
United Aircraft Research Laboratories SSS 10	1.86	0.31	1.03	Ref. 3-21 Calculated values over simulated Federal Driving Cycle, based on emission data from GM engine T-56.
Chrysler sixth-generation engine	3.99	-0.26	2.21	Ref. 3-31, 1975 Federal Test Procedure emissions corrected (background level subtracted from measured values).
Williams Research/ WR26/AMC Hornet	7.43 6.92	0.62 0.72	2.8 2.5	Cold start Hot start } Ref. 3-19 test data, 1975 Federal Test Procedure
Williams Research/ 131Q/Volkswagen	4.5	0.34	1.81	Ref. 3-20 test data, 1972 Federal Test Procedure.
b. Advanced Combustors				
Solar	3.34	0.11	0.34	Calculated values for simulated Federal Driving Cycle, based on advanced combustor data. (Ref. 3-23)
General Motors 225-horsepower regenerative engine	2.4	0.015	0.315	Experimental test bed engine, chassis dynamometer test with 5,000-lb car, 1975 Federal Test Procedure (Ref. 3-18).
AiResearch	1.6	0.67	0.44	Calculated values for simulated Federal Driving Cycle, for recuperated engine with 10 percent bypass. (Ref. 3-25)
United Aircraft of Canada	3.64	0.49	0.52	Calculated values over simulated Federal Driving Cycle based on advanced combustor data. (Ref. 3-29)
c. Clean Air Act Requirements	3.4	0.41	0.4	1976 Federal standards

Table 3-4. The exhaust emissions are generally close to or below the original 1976 requirements. The AiResearch data, which simulates the use of an advanced combustor in a recuperative engine with 10 percent recuperator bypass, meets the CO and HC standards but is slightly above the NO_x requirement. These laboratory tests have proven the feasibility of reducing exhaust emissions to meet the 1976 requirements.

A program sponsored by General Motors has resulted in installing an experimental 225-horsepower gas turbine engine in a car meeting the original 1976 Federal emission standards. This workhorse two-shaft engine has single regenerator disc; it operates at a maximum gasifier (compressor-turbine speed of 44,000 rpm, 4.5 atmospheres pressure ratio, 1850°F maximum turbine inlet temperature and specific fuel consumption of 0.55 lb/bhp-hr. The test was conducted with an external control system that has not yet been developed into an on-board system (Ref. 3-32).

3.3.2 Fuel Economy

The estimated SFC for design point operation of several engines was tabulated in Table 3-1; design SFC values ranging from 0.4 to 0.95 were shown. However, since automobiles operate over a range of part-load conditions, the performance indicated at the design point is only crudely representative of engine fuel economy over a normal driving cycle.

Fuel economy can be evaluated more realistically when engine operation is simulated for an automobile run over the FDC. In the engine optimization studies performed for EPA, the fuel economy of several gas turbine engines was evaluated and compared with the fuel economy of a representative V-8 spark-ignition engine.

Fuel economy values estimated by United Aircraft for single-shaft, 150-hp gas turbine vehicles are shown and compared with a baseline Otto cycle vehicle (OC-70) in Table 3-5. The major conclusion drawn from this comparison was that the fuel economy of a regenerative or recuperative engine was superior to that of a simple-cycle gas turbine engine or a spark ignition engine.

Table 3-5. United Aircraft Estimated Fuel Economy, Baseline Vehicles (Ref. 3-21).

Ambient Temperature 59°F

Fuel Economy (mpg)								
Engine	FDC	20 mph	30 mph	40 mph	50 mph	60 mph	70 mph	Vehicle Test* Weight (lb)
RGSS-6 (Regenerative Single Shaft)	12.16	15.06	18.02	18.65	17.96	16.59	14.70	4000
RCSS-8 (Recuperative Single Shaft)	11.20	13.73	16.59	17.33	16.73	15.42	13.79	3950
SSS-10 (Simple Cycle)	7.51	8.86	11.21	12.51	12.63	12.22	11.47	3700
OC-70**	9.49	10.65	12.62	14.10	13.71	12.91	11.48	4300

*Simulated
**Otto Cycle

AiResearch predictions of fuel economy are shown in Table 3-6. Calculated results for two free-turbine, regenerated cycle, 135-hp and 175-hp engines, operating with three-speed automatic transmissions, are compared with a spark ignition engine. Both FDC and fixed operating speed results are indicated. The fuel economy over the FDC is highest for the small

Table 3-6. AiResearch Estimated Fuel Economy of Free-Turbine Regenerated Cycle (4000-lb Test Weight Vehicle,* 4-hp Accessory Load, Three-Speed Automatic Transmission), (Ref. 3-1).

Rated Power Level at 59°F Without Boost, shp	Fuel Economy, mpg							
	Ambient Temperature, °F	FDC (1.3-hp Accessory Load)	20 mph	30 mph	40 mph	50 mph	60 mph	70 mph
175	85	16.4	20.6	23.2	23.3	23.4	22.3	20.6
135	85	18.1	23.3	23.8	25.9	25.8	24.3	21.1
175	59	12.5	16.1	18.2	18.2	17.2	15.5	13.7

*Simulated

135-hp engine. This engine requires water injection to meet the maximum power requirements. The fuel economy of the 175-hp regenerative engine, while less than that of the 135-hp engine, shows a 31-percent improvement over the spark ignition engine.

Existing gas turbine automobiles, however, show lower fuel economy. The average fuel economy of the Chrysler sixth-generation gas turbine engine installed in the intermediate-size car is approximately 8.5 miles per gallon over the FDC. The peak fuel economy of the same car is 18 miles per gallon at 40 miles per hour (Ref. 3-17). The AMC Hornet with the Williams Research WR-26 engine rated at 80 horsepower had a fuel economy performance range of 5.1 to 8.1 miles per gallon during several tests conducted for exhaust evaluation using the FDC (Ref. 3-19).

3.3.3 Engine Noise and Vibration

The noise emanating from gas turbine engines is composed of different frequencies with the spectrum varying with engine speed. The noise level tends to be higher at idle speeds and lower at high speeds, when compared to an internal combustion engine, because of the high-velocity flow of air at the compressor inlet, high idle speeds, and interference effects due to the variable inlet guide vanes. The gas turbine engine also lacks the damping characteristics provided by the concentrated mass of the internal combustion engine block. In general, the regenerative gas turbine engine may be expected to have a lower noise level than the simple-cycle engine, because the heat exchanger in the regenerative case absorbs part of the sound energy in the exhaust.

A number of investigators expect that the automotive gas turbine engine system will require exhaust-noise silencers (particularly for simple-cycle engines). The present Chrysler engine package is reported not to require exhaust mufflers.

Chrysler expects that a number of design modifications would be required in the gas turbine vehicle's chassis and frame. These include a new suspension system, a new exhaust system arrangement, and a new front end. The new suspension would be of the rubber-insulated type to provide a quieter ride.

According to drivers in the Chrysler 50-car test program (Ref. 3-22), the gas turbine engine operated with significantly less vibration than the conventional piston engine. This is directly attributable to the pure rotary motion of the engine.

3.3.4 Maintenance and Durability

Modern piston engines operated in normal passenger car service as expected to accumulate 75,000 to 100,000 miles without requiring a major overhaul. The impact of exhaust emission controls on this type of durability (and the expected reliability) is still under evaluation. The gas turbine engine, with fewer parts may, assuming proper care and maintenance, provide even higher durability.

Contractors involved in the EPA-sponsored engine configuration optimization study addressed the problem of gas turbine car maintenance and concluded that scheduled maintenance and service requirements (chassis lubrication, air and oil filter changes, etc.) would be much the same as for the conventional automobile. AiResearch, for example, estimated that engine-related repairs, although different in nature, would occur at about the same frequency among the aggregate of service or repair items for both engine types.

In the Chrysler 50-car program (Ref. 3-22), 1.1 million vehicle miles were accumulated in a period slightly over 2 years. In the beginning of this program, the lost time due to engine malfunction was about 4 percent; this was eventually reduced to 1 percent as familiarity with repair problems was acquired. The Chrysler experience suggests that the training of mechanics in maintenance and repair of gas turbines will not present unusual problems.

Chrysler anticipates that production models of gas turbine engines will require less maintenance than their internal combustion engine counterparts. This judgment is partly based upon experience with the sixth-generation prototype engine, which was serviced after 3,500 hours of operating time, corresponding to a mileage accumulation of about 170,000 miles.

A potentially serious service problem in regenerated gas turbines concerns the high frequency of maintenance encountered with the regenerator seal, a seal problem that must be solved before this engine type can be placed into mass production.

3.3.5 Safety

Aircraft and stationary gas turbine engines have an excellent record of safety performance, due largely to the practice of employing regularly scheduled preventive maintenance and overhaul. In the passenger car application, systematic maintenance cannot be guaranteed. Accordingly, special safety provisions must be incorporated into the system's basic design to prevent severe damage in the event of catastrophic failure, as follows:

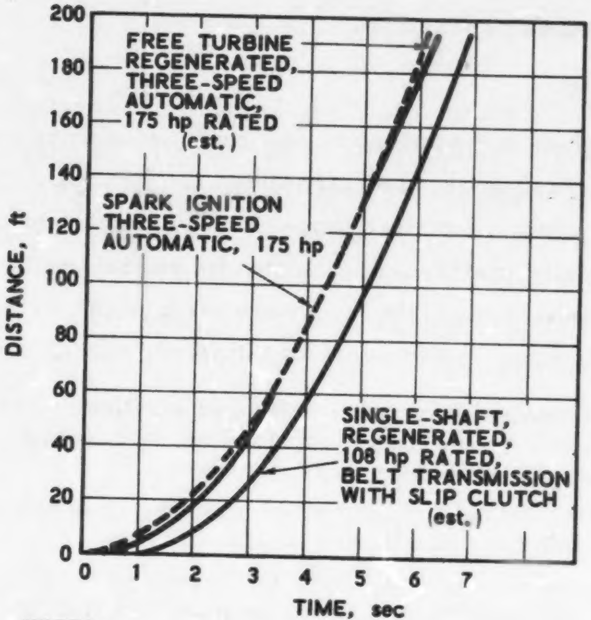
- Containment has to be provided so that in case of engine runaway or turbine blade failure, the rotating elements of the engine will be safely confined.
- Limit controls must be designed so that engine operation will not exceed allowable limits.
- Backup provisions, such as engine packaging and installation arrangements, must be made so that the vehicle occupants are not subjected to heat or other hazards in the event that primary safety provisions fail under high-impact accident or other catastrophic conditions.

No fundamental difficulties are anticipated in meeting these objectives for the passenger car turbine.

3.3.6 Driveability

In general, the gas turbine car does not have the standing-start response of a piston-powered vehicle, circa 1970. This is exemplified by the performance characteristics shown in Figure 3-16 where the standing-start acceleration performance for a regenerated free turbine engine vehicle and a single-shaft turbine engine vehicle are compared with a spark ignition engine vehicle. As shown, it is estimated that the single-shaft system has up to a 1-second delay in starting vehicle motion, and the free turbine has up to a 0.4-second starting delay; the precontrolled emission, spark ignition engine has a delay of less than 0.1 second.

Some comments on driveability by drivers participating in Chrysler's 50-car test program, as noted in Reference 3-22, are as follows: (1) The operation of the engine was smooth and vibrationless. Passengers



NOTES:

1. T_{AMB} = 105°F
2. VEHICLE WEIGHT = 4000-lb TURBINE ENGINE
   = 4300-lb SI ENGINE

Figure 3-16. Standing-Start Acceleration Performance (Ref. 3-1)

felt a gliding sensation at all speeds, which was particularly pleasant on long trips. (2) Starting ability was said to be superior to that of the internal combustion engine. Users consistently stated that the turbine car was superior to conventional cars in providing fast and sure ignition. (3) Other superior operating features mentioned were good engine power, low vibration, and nonstalling characteristics. (4) Users tended to contradict each other in reaction to the sound of the turbine engine. For every person who complained about the noise level, there were three or four who liked the sound. (5) The main negative comment was lack of acceleration, primarily when starting from standstill. Chrysler attributed this to the relatively low engine design power level in this automobile (130-hp).

Chrysler improved the acceleration and braking response characteristics of their sixth-generation engine, relative to the fourth-generation engine performance, by shifting major accessories from the first-stage turbine shaft to the power turbine shaft. Braking response characteristics were improved with an increase in turbine inlet temperature, providing additional braking power at negative turbine nozzle blade incidence. These improved characteristics were displayed in Figure 3-14.

3.3.7 Starting Characteristics

The time required from driver initiated engine startup until power is available for vehicle acceleration is a performance factor. The cold-engine, start-time characteristics of the Chrysler sixth-generation automobile gas turbine (see Figure 3-17) show that starting time is significantly affected by the ambient temperature.

3.4 CURRENT STATUS OF TECHNOLOGY

3.4.1 Current Utilization

As pointed out before there are no gas turbine automobiles in production today. The only engine considered as a possible candidate for production was the Chrysler engine in their well publicized 50-car program. The engine specification and design point characteristics are shown in Table 3-7 (Ref. 3-22).

During the Chrysler 50-car, gas-turbine test program, which lasted for 2 years (October 1963 to January 1966), cars were assigned to selected individuals and, as reported by Chrysler, the reaction was generally favorable. However, in 1966 Chrysler decided to abandon their original idea of a low-volume-production, gas turbine vehicle because of a number of deficiencies uncovered during the program. These included poor fuel economy, inadequate engine response and braking, insufficient engine power, and excessive noise. They have continued gas turbine work as a research program, as described in the next section.

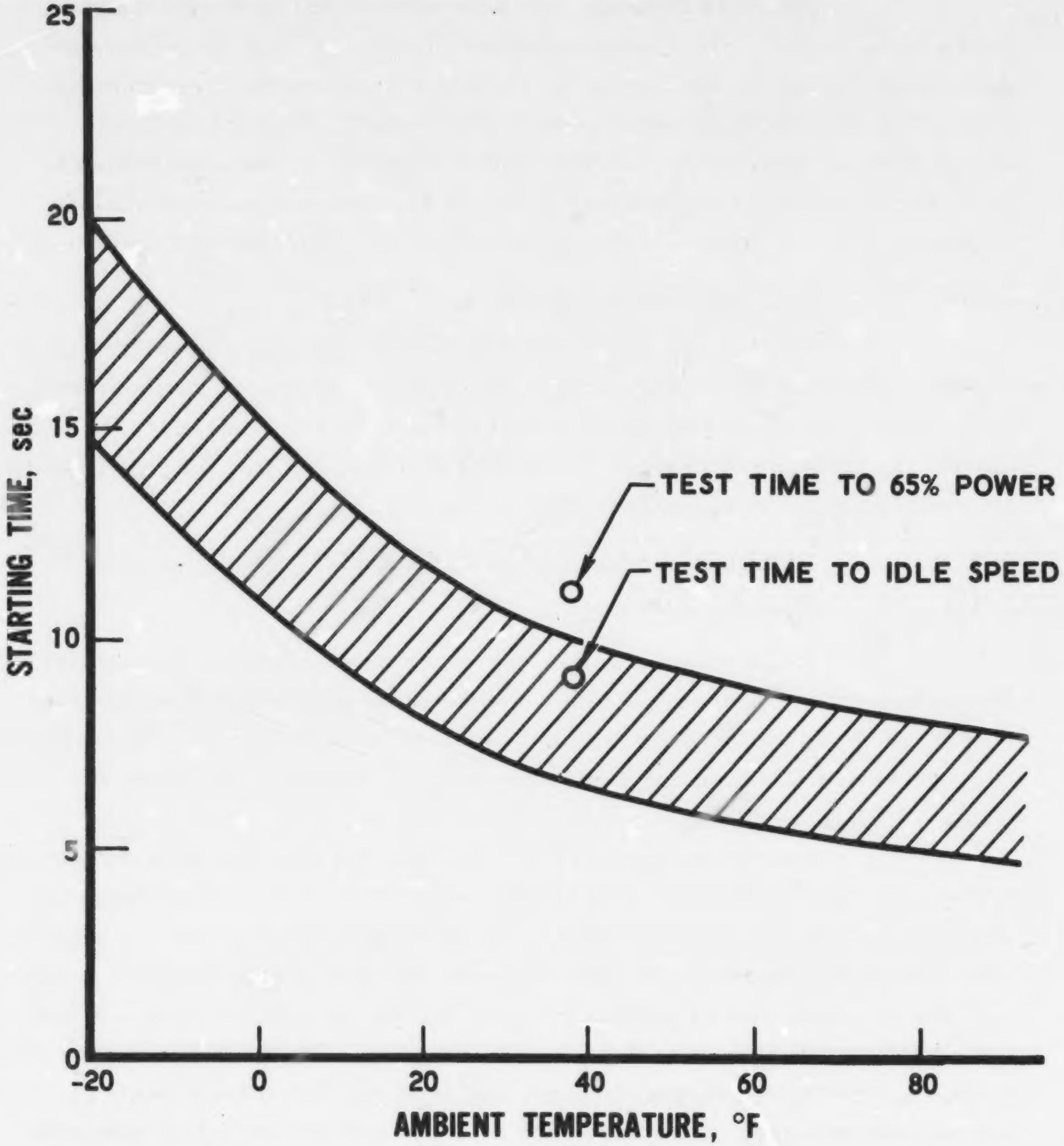


Figure 3-17. Starting Time versus Ambient Temperature (Ref. 3-17)

Table 3-7. Specifications of Chrysler Corporation's Fourth-Generation Gas Turbine Engine (Ref. 3-22)

General

Type:

Regenerative gas turbine

Rated Output:

Power - 130 bhp @ 3,600-rpm output shaft speed
Torque - 425 lb-ft @ zero-rpm output shaft speed

Weight:

410 lb

Basic Engine Dimensions (without accessories):

Length - 25 inches
Width - 25.5 inches
Height - 27.5 inches

With automotive accessories in place, the overall length is:

35 inches

Fuels:

Unleaded gasoline, diesel fuel, kerosene, JP-4, etc.

Components

Compressor:

Type - Centrifugal
Stages - One
Pressure Ratio - 4:1
Efficiency - 80%

Table 3-7 (continued)

First Stage Turbine:

Type - Axial
Stages - One
Efficiency - 87%

Second Stage Turbine:

Type - Axial
Stages - One
Efficiency - 84%

Regenerator:

Type - Two rotating disks
Effectiveness - 90% +

Burner:

Type - single can, reverse flow
Efficiency - 95%

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*DESIGN POINT CHARACTERISTICS

Maximum Gas Generator Speed - 44,600 rpm
Maximum Second Stage Turbine Speed - 45,700 rpm
Maximum Output Speed (after reduction gears) - 4,680 rpm
Maximum Regenerator Speed - 22 rpm
Compressor Air Flow - 2.2 lb/sec
First Stage Turbine Inlet Temperature - 1700°F
Exhaust Temperature (full power) - 525°F

*Ambient conditions:

Temperature - 85°F
Barometric Pressure - 29.92 in. Hg

3.4.2 Current Research and Development

3.4.2.1 Engine Research and Development

The main EPA-sponsored engine research program is with the Chrysler Corporation.

Research and development on the automotive gas turbine has been under way at Chrysler since 1950. The current engine is a modification of their fourth-generation engine used in their 50-car test program. The engine is a free-turbine regenerative design consisting of the following basic components: (1) a single-stage radial compressor (4.1:1 design pressure ratio, 44,600-rpm design speed) driven by a single-stage axial turbine; (2) two rotating regenerators; and (3) a separate variable nozzle, single-stage axial power turbine (45,700-rpm design speed). The engine is connected to the vehicle transmission through a reduction gear arrangement. Axial turbine wheels were selected by Chrysler for a number of reasons, including improved packaging and low inertia.

The principal new features of the current engine are a higher first-stage turbine inlet temperature--1,850°F vs 1,750°F during maximum steady-state operation; 2,000°F vs 1,850°F during maximum acceleration, a higher power output (150-hp vs 130 hp)--and a redesigned accessory drive arrangement. On this current engine, major accessories (power steering and alternator) are driven by the power turbine. In the fourth-generation engine, all accessories were driven by the gas generator. The shift of major accessories to the power turbine resulted in a faster response for the engine, as shown in Figure 3-17. This change reduced the first-stage turbine load, permitting a reduction in the size and number of high-speed bearings required.

This 150-hp, sixth-generation engine, shown schematically in Figure 3-18 and pictorially in Figure 3-19, is a research and not a production engine. The fuel economy and the power output of the engine were increased from the fourth-generation values by increasing the first-stage turbine inlet temperature by 100°F. This was accomplished by fabricating the turbine

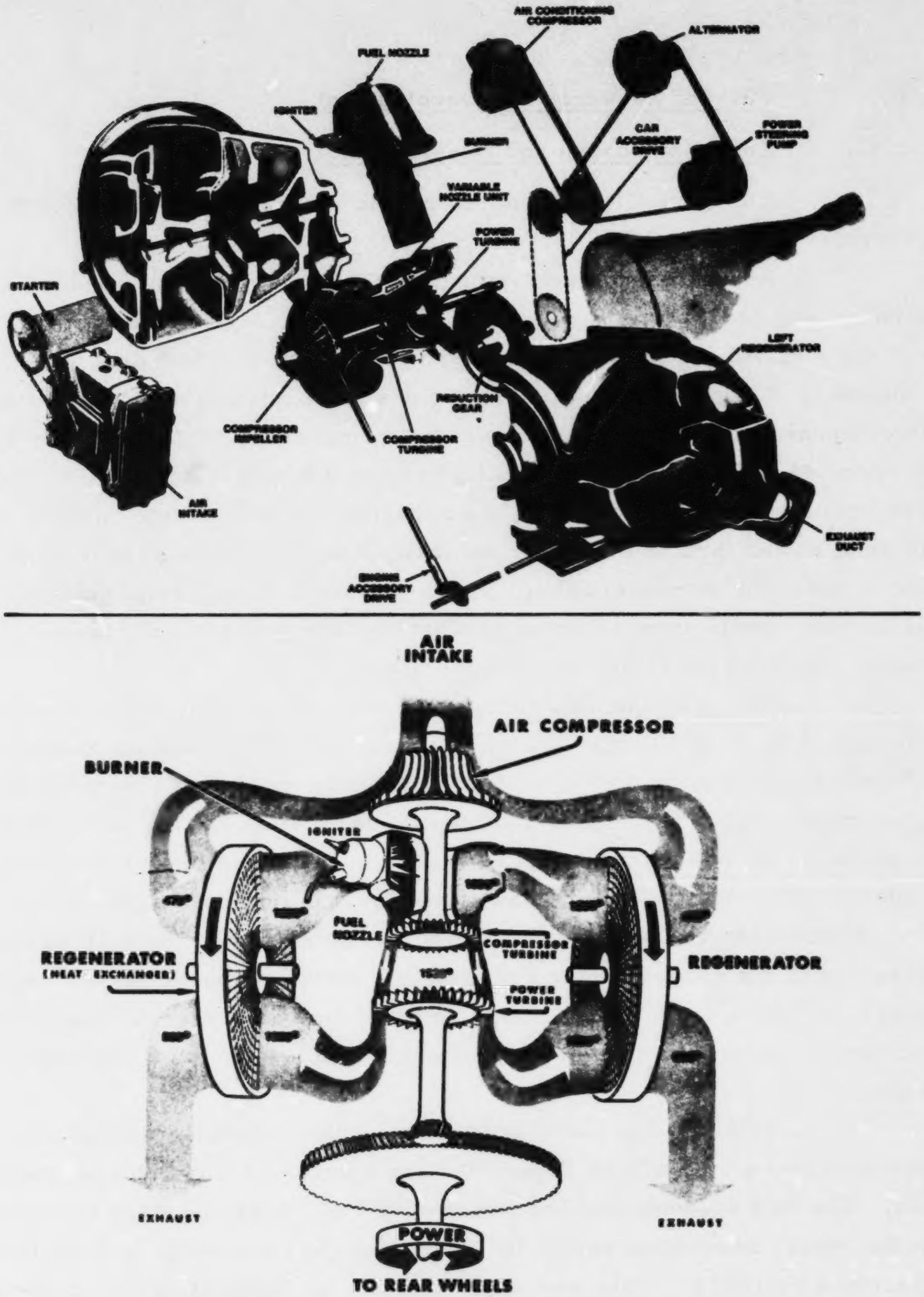

AIR CONDITIONING COMPRESSOR
ALTERNATOR
FUEL NOZZLE
IGNITER
BURNER
CAR ACCESSORY DRIVE
POWER STEERING PUMP
VARIABLE NOZZLE UNIT
POWER TURBINE
STARTER
LEFT REGENERATOR
REDUCTION GEAR
COMPRESSOR IMPELLER
COMPRESSOR TURBINE
AIR INTAKE
EXHAUST DUCT
ENGINE ACCESSORY DRIVE
AIR INTAKE
AIR COMPRESSOR
BURNER
IGNITER
FUEL NOZZLE
COMPRESSOR TURBINE
POWER TURBINE
REGENERATOR (HEAT EXCHANGER)
REGENERATOR
EXHAUST
EXHAUST
POWER
TO REAR WHEELS

Figure 3-18. Schematic of Chrysler Sixth-Generation, Gas Turbine Engine (Ref. 3-17)

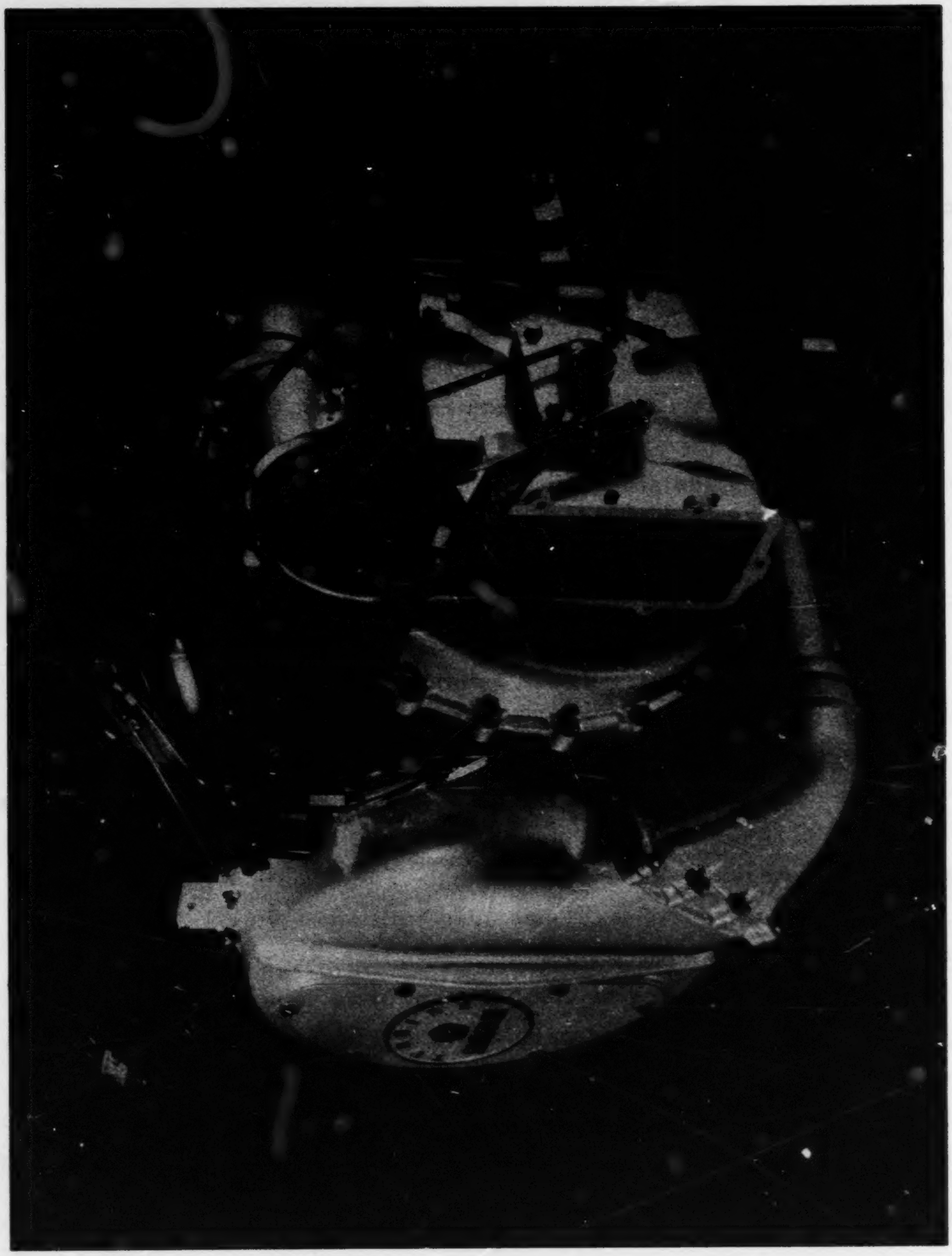


Figure 3-19. Chrysler 150-hp Gas Turbine Engine (Ref. 3-17)

wheels from INCO 713C (70-percent nickel). Additional illustrations of Chrysler's gas turbine engine installed in automobiles are given in Figures 3-20 through 3-23.

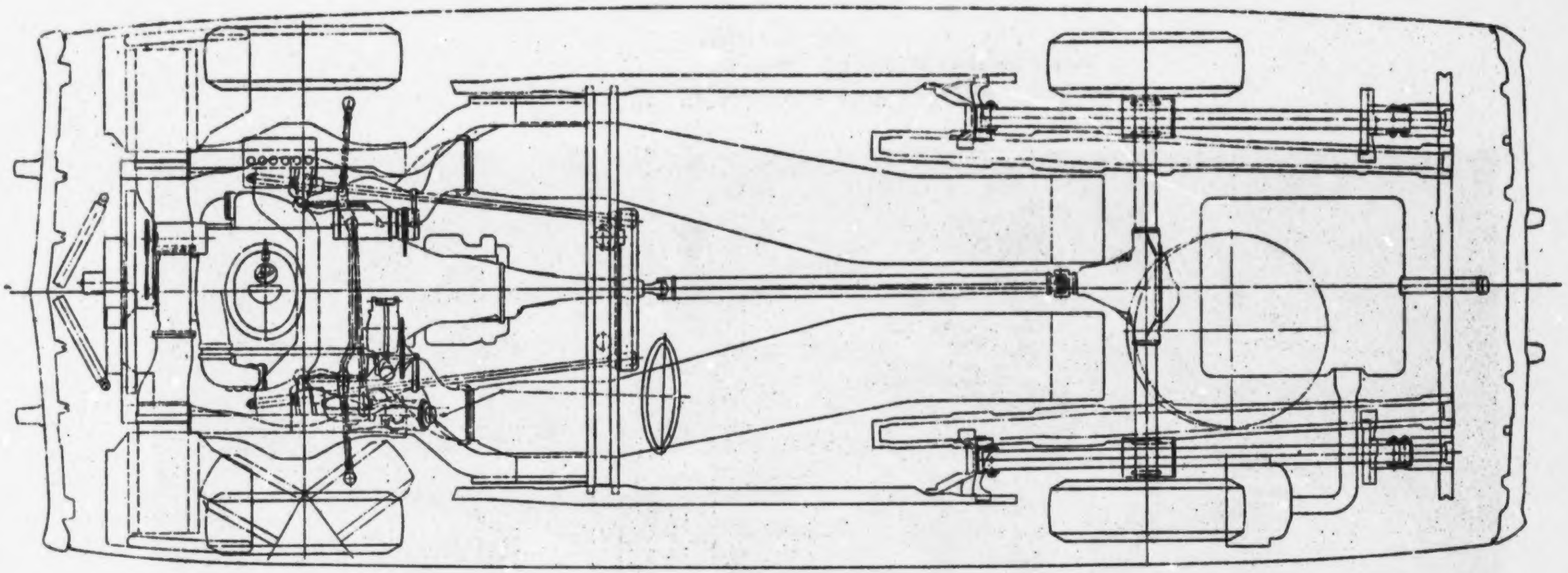
In December 1972, Chrysler was awarded a $6.5 million contract by EPA for the development of an improved version of the sixth-generation gas turbine engine with respect to emissions and fuel economy. Tests conducted to date with the 1972 Federal Test Procedure indicate that the original 1975 Federal emission standards can be met by the sixth-generation engine. However, NO_x emissions of 2.7 gm/mile obtained on this engine are considerably above the requirement of 0.4 gm/mile. Reduction of NO_x is sought primarily by means of combustor modifications. Both in-house combustor designs and combustors designed under separate EPA contracts will be tested by Chrysler. The NASA Lewis Research Center is playing an important role in the effort in the areas of combustion, rotating machinery design, and engine test.

In addition to the U.S. research and development described above, a number of foreign activities in automotive gas turbine development are in progress. Volkswagen, for example, has undertaken a long-range program to develop a passenger car gas turbine with performance and production costs comparable to the spark ignition engine but with improved exhaust emission characteristics. This program is in the preprototype hardware development stage. A research engine developed with Williams Research Corporation, the VW-GT 70, has been built and tested. This is a two-shaft, 75-hp machine with preheating and an integral reduction gear designed to fit into the rear engine compartment of various Volkswagen vehicles. Emissions over the Federal Driving Cycle for the VW-GT 70 are reported to meet the original U.S. standards for 1975; however, NO_x emissions are still significantly higher than the 1976 0.4 gm/mi level.

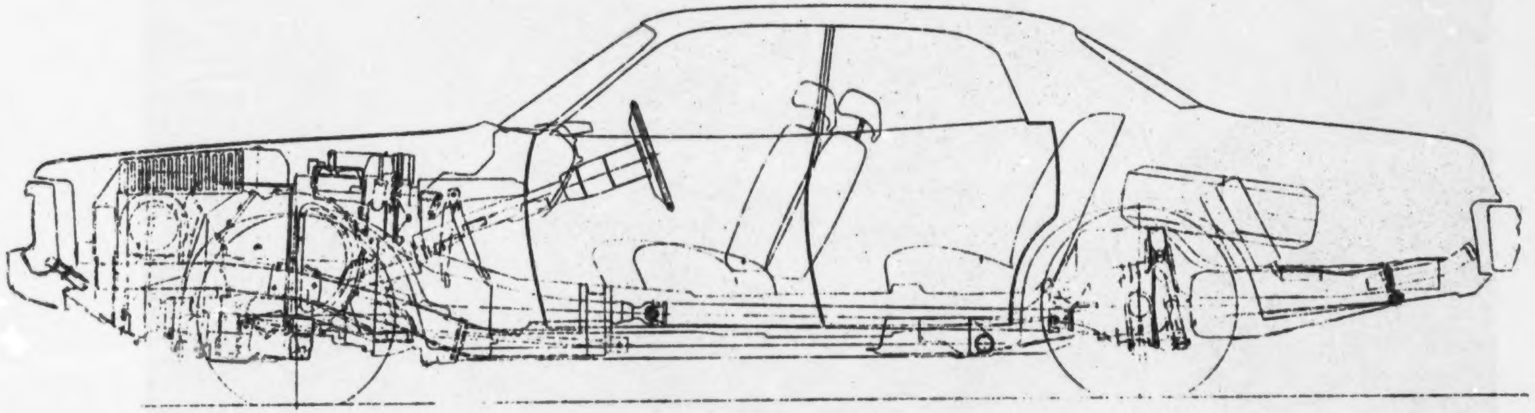
3.4.2.2 Components Research

3.4.2.2.1 Combustors

Development work conducted on gas turbine combustors has indicated that sufficiently low HC and CO emission levels can b[illegible]ved



TOP VIEW



SIDE VIEW

Figure 3-20. Chrysler Baseline Turbine-Powered Vehicle (Ref. 3-30)

Figure 3-21. Chrysler 150-hp Gas Turbine Engine, Vehicle Installation (Ref. 3-30)

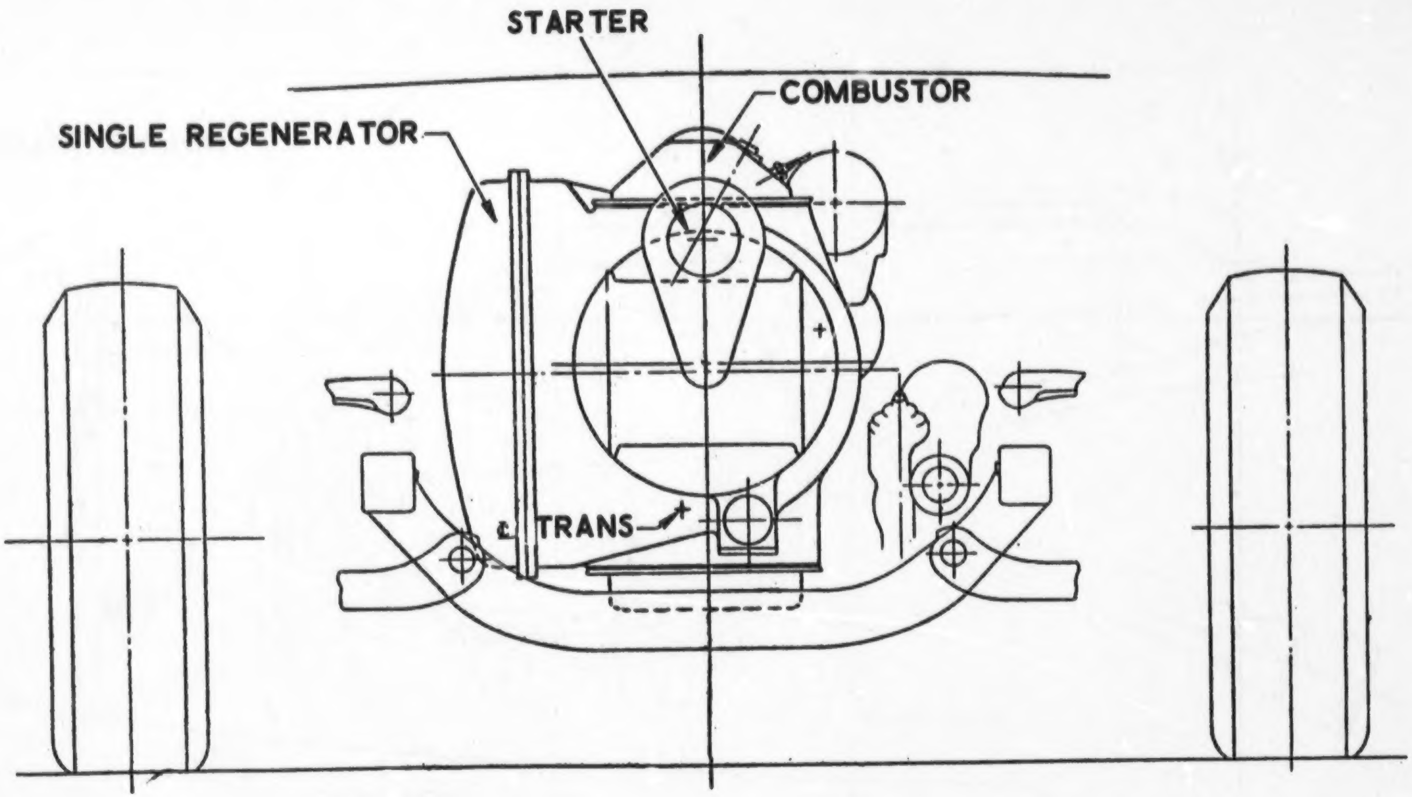

STARTER
COMBUSTOR
SINGLE REGENERATOR
℄ TRANS

Figure 3-22. Chrysler 100-hp Upgraded Gas Turbine Engine with Single Vertical Regenerator Compact Vehicle Installation (Ref. 3-30)

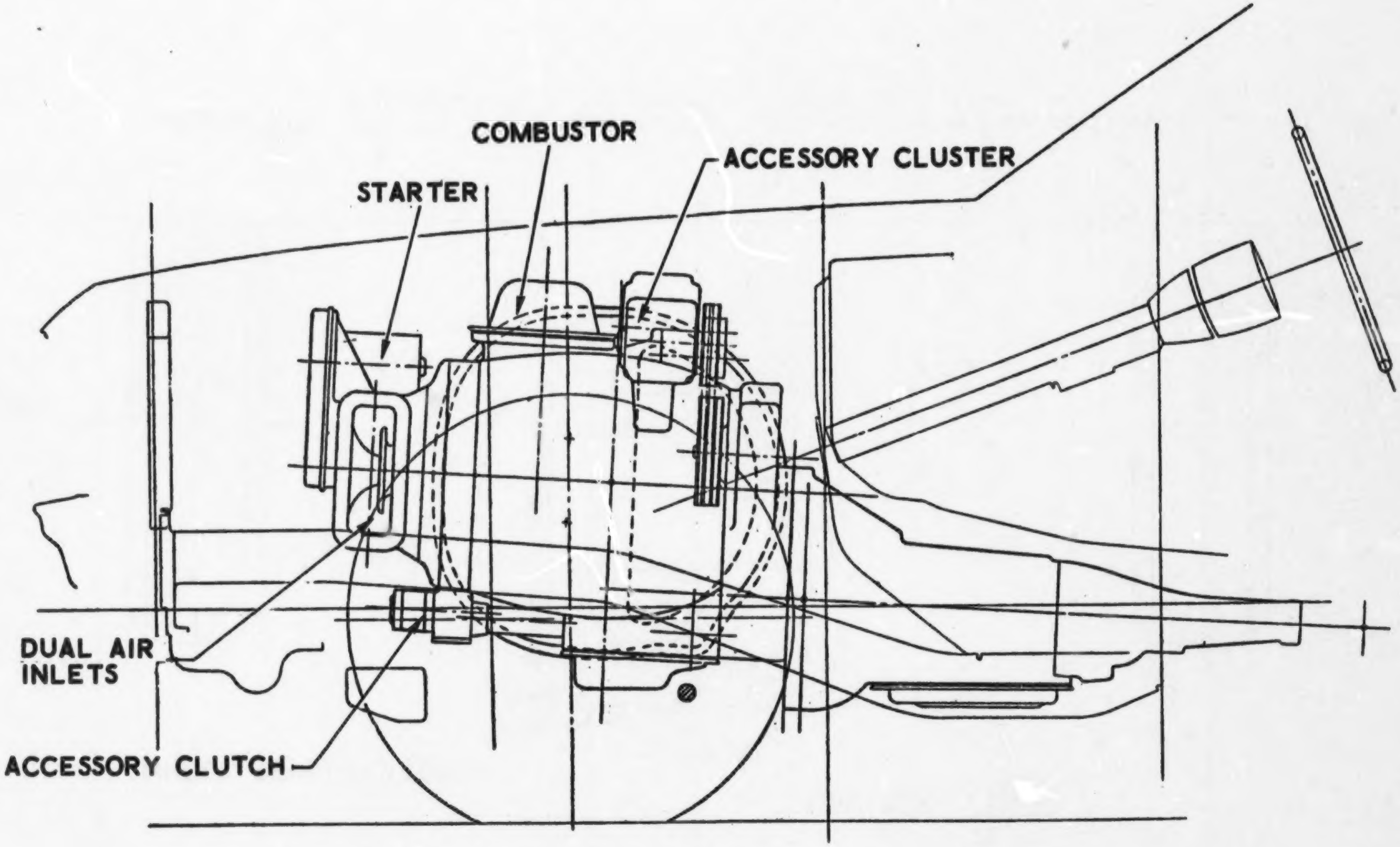

COMBUSTOR
ACCESSORY CLUSTER
STARTER
DUAL AIR INLETS
ACCESSORY CLUTCH

Figure 3-23. Chrysler 100-hp Upgraded Gas-Turbine Engine with Single-Shaft Vertical Regenerator Compact Vehicle Installation (Ref. 3-30)

with conventional combustors to meet the federal emission control objectives for light duty vehicles. However, the levels of NO_x emitted by these combustors are considerably higher than the original 1976 0.4 gm/mi requirement. In conventional combustors, the fuel is injected at high pressure through fuel injectors, and fuel atomization is achieved in the dome section of the cylindrical or annular combustion chambers by means of jet impingement. Combustion of the fuel takes place in the near-stoichiometric fuel vapor/air region surrounding the individual fuel droplets. As a result, the combustion temperatures are very high, especially in the case of regenerated or recuperated gas turbine engines. Since the kinetically controlled NO_x formation rate increases exponentially with temperature, the NO_x emissions from a conventional droplet-burning-type combustor are also very high. For instance, Chrysler's sixth-generation gas turbine installed in a medium-size vehicle has NO_x emissions of about 2.2 gm/mi over the Federal Driving Cycle. The NO_x emissions of nonregenerated gas turbines with conventional combustors tend to be somewhat lower, because of the lower temperature of the incoming air.

In order to obtain satisfactory fuel economy in gas turbines, some degree of regeneration appears to be required. Since conventional gas turbine combustors are inherently high NO_x emitters it appears that new combustors would have to be developed if the gas turbine automobile is to meet federal NO_x objectives. In view of current industry efforts in the development of high-temperature gas turbines utilizing ceramic components, the need for low NO_x combustors becomes even more important.

The concept of lean combustion of fully vaporized and premixed air fuel mixtures offers a potential solution to the NO_x emission problem with current gas turbine engines. New combustor designs based on this particular concept are now in the development stages. Although considerable progress has been made to date by industry, much more remains to be done, particularly in the area of durability and off-design operation. A number of these advanced configurations are briefly described in the following paragraphs.

Solar,[*] under contract to the EPA, has been involved in the development of a low NO_x, jet-induced circulation (JIC) combustor concept for potential use in automotive gas turbines (Figure 3-24). In this design, the fuel and the primary zone air are premixed before entering the primary combustion zone through a variable nozzle-like opening. Emission levels obtained during testing over a simulated Federal Emission Test Driving Cycle indicate the feasibility of meeting one-half of the original 1976 Federal Emissions Standards. Further development of the combustor and nozzle activator mechanism is needed before a combustor/engine combination could be used to power a vehicle.

In the Ford external vaporizing combustion (EVC) concept, the fuel enters the premix/vaporization chamber through a coaxial nozzle and is then atomized and premixed by the high-velocity primary air flow (Refs. 3-4 and 3-5). Based on initial test results, Ford feels that the emission control objectives for light-duty vehicles can be met with a gas turbine utilizing the lean premixed/prevaporized EVC concept. However, a number of potential problem areas must be solved before the combustor is ready for use. These include combustor stability and response under transient operating conditions, preignition, durability, and cold-start emission characteristics.

Aerojet has designed a novel combustor for potential use in the EPA automotive gas turbine program. In this concept (Ref. 3-24) the conventional fuel-spray nozzles are replaced by the premixer hardware which distributes the initially liquid fuel, then injects it into the combustor air through a large number of very small orifices (Figure 3-25). Based on analytical work on flow mixing and combustion kinetics, Aerojet feels that emission control requirements can be met with the platelet combustor.

AiResearch, under contract to EPA, has been involved in the development of low-NO_x combustors utilizing recuperator bypass (Ref. 3-25). A portion of the recuperator bypass air is injected through the dome section

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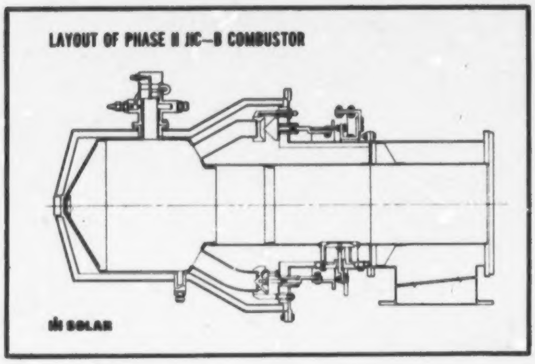
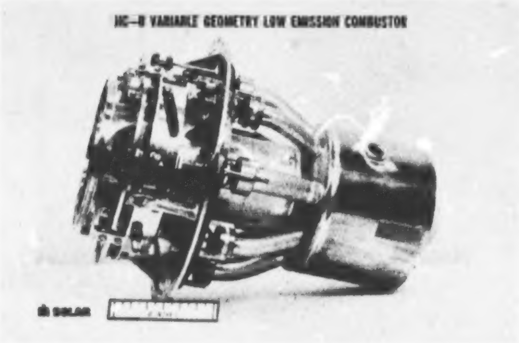
[*] Solar Division of International Harvester Company


LAYOUT OF PHASE II JIC—B COMBUSTOR
SOLAR


JIC—B VARIABLE GEOMETRY LOW EMISSION COMBUSTOR
SOLAR

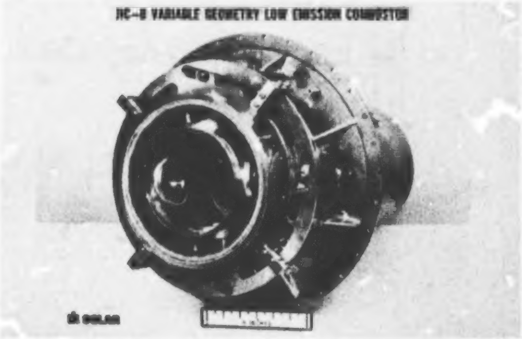

JIC—B VARIABLE GEOMETRY LOW EMISSION COMBUSTOR
SOLAR

Figure 3-24. Solar Jet-Induced Circulation Combustor, JIC-B(Ref. 3-23)

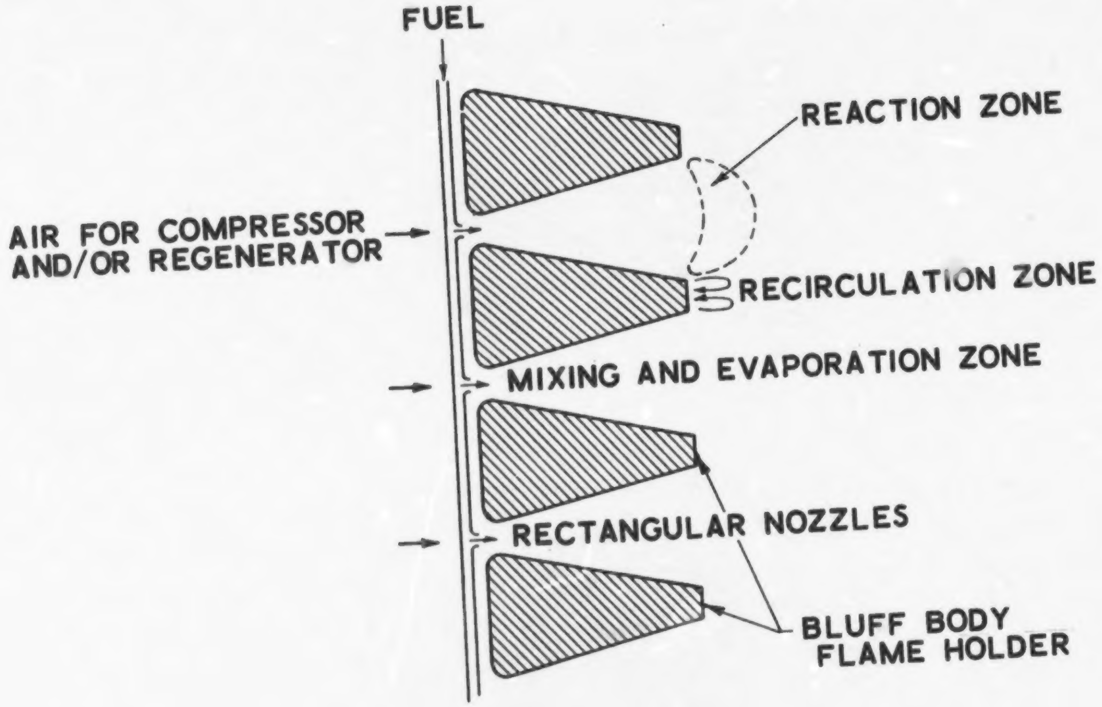

FUEL
REACTION ZONE
AIR FOR COMPRESSOR AND/OR REGENERATOR
RECIRCULATION ZONE
MIXING AND EVAPORATION ZONE
RECTANGULAR NOZZLES
BLUFF BODY FLAME HOLDER

Figure 3-25. Schematic of Aerojet Platelet Premixer Concept (Ref. 3-24)

of the combustor, the remainder of the bypass air and fuel are injected in an upstream direction through an L-shaped pipe located downstream of the dome (Figure 3-26). Although the emissions of this vaporizer combustor with bypass are rather low, further reduction of NO_x is required before the federal standards can be met in a gas turbine automobile.

In another EPA contract effort, General Electric worked on the development of a low-emission porous plate combustor for use in automotive gas turbine engines. In this concept (Ref. 3-26), the fuel is prevaporized and the air fuel mixture is then discharged through the porous combustor plate and ignited at the front face of the plate (Figure 3-27). Emission measurements with propane/air mixtures were reported to be favorable (Ref. 3-26), with NO_x and HC generally below the original 1976 Federal emission standards. CO was below the standards at combustor conditions corresponding to light engine loads but above the standards at heavier loads.

In catalytic combustors a catalyst material, deposited on a suitable ceramic substrate, is used to promote and sustain the oxidation reactions at very lean air fuel ratios. Preliminary test data indicate catalytic combustors are inherently low NO_x, CO, and HC emitters. However many potential problems remain to be solved. These include catalyst and substrate life, catalyst poisoning characteristics, temperature capability, transient response, maximum heat-release rate, and air fuel mixture preparation.

3.4.2.2.2 Ceramic Components

Fuel consumption of gas turbine engines decreases as turbine inlet temperature is increased. The specific fuel consumption of a highly regenerated engine operating at a compressor-pressure ratio of from 4:1 to 6:1 can be improved by about 20 percent by increasing the turbine inlet temperature from 1,800°F to 2,500°F. Based on current materials technology, turbine inlet temperatures are limited to about 1,800°F to 1,900°F using nickel alloys. Higher temperatures can be achieved with transpiration-cooled, veil-cooled, or internally cooled turbine blades. However, these

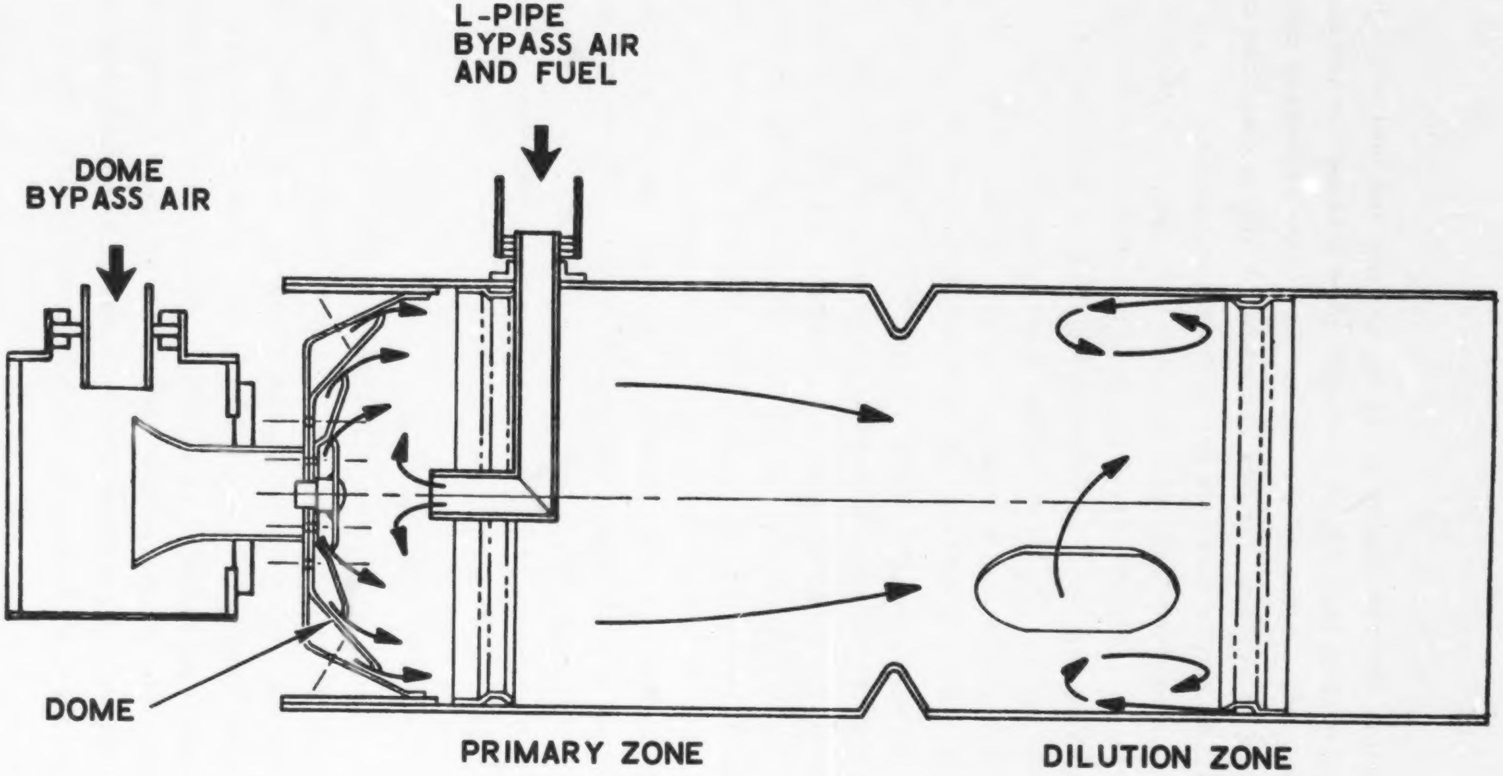

L-PIPE
BYPASS AIR
AND FUEL
DOME
BYPASS AIR
DOME
PRIMARY ZONE
DILUTION ZONE

Figure 3-26. AiResearch Recuperator Bypass Combustor (Ref. 3-25)

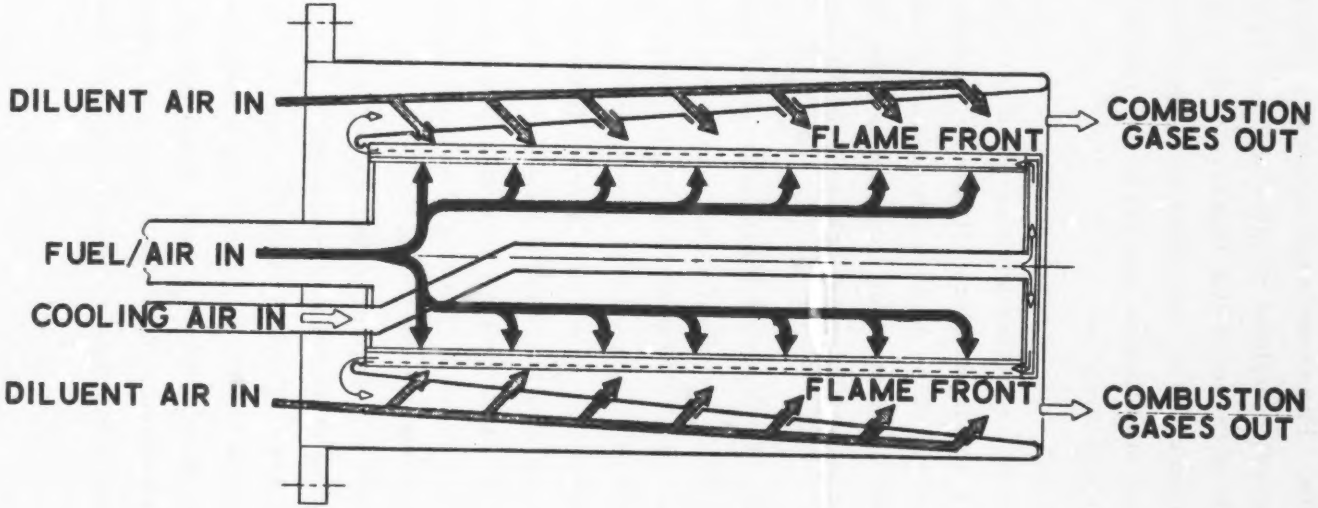

DILUENT AIR IN
FUEL/AIR IN
COOLING AIR IN
DILUENT AIR IN
FLAME FRONT
FLAME FRONT
COMBUSTION GASES OUT
COMBUSTION GASES OUT

Figure 3-27. General Electric Low-NO_x Porous-Plate Gas Turbine Combustor (Ref. 3-26)

approaches are limited to turbine inlet temperatures of about 2,300°F and are not currently considered feasible for automotive gas turbines because of high manufacturing costs. For these reasons, gas turbine manufacturers have been interested in ceramic turbine nozzles and rotors for some time. Although initial acceptance of these brittle materials has not been good, there are indications that the properties of ceramics have now been improved to a point where a number of manufacturers are seriously interested in the use of these materials in their gas turbines.

No serious difficulties are foreseen in the use of ceramics for stationary components of the engine such as combustor liners, nozzles, shrouds, heat exchangers, and insulation. However, the design of rotating components such as ceramic turbine wheels represents a very difficult challenge. The wheel must be capable of withstanding the high thermal and centrifugal loads and must be sufficiently light to provide adequate engine response.

In June 1971, Ford, with Westinghouse as subcontractor, was awarded a $10.3 million ARPA-sponsored contract to demonstrate that brittle materials can be successfully used in high-temperature gas turbine components such as turbine inlet nose cones, nozzle vanes, turbine wheels, and combustor liners. Under terms of the contract (which is monitored for ARPA by the Army Materials and Mechanics Research Center) Ford is concentrating on development of ceramics for use in a small automotive gas turbine vehicle, and Westinghouse is investigating the applicability of ceramics to large stationary gas turbines (Ref. 3-27).

To date, Ford has concentrated its efforts in the areas of materials technology, turbine component development, and stress analysis. In the material technology area, Ford initiated a program to determine the physical properties of various ceramics, including hot-pressed silicon carbide, hot-pressed silicone nitride (Si_3N_4), and injection-molded silicon nitride. Based on these evaluations, Ford feels that hot-pressed silicon nitride and silicon carbide are candidate materials for turbine rotors.

Although very encouraging results have been achieved to date in the ceramics area, considerably more development work is required before these ceramics can be considered feasible for use in the manufacture of rotating components for applications to the automotive gas turbine engine. More data are needed on material behavior as a function of time, temperature, and method of fabrication. The effect of fuel impurities (e.g., sulfur) should be evaluated. Standardization of property testing and reporting is needed. Product uniformity must be developed. The designer will have to learn to cope with the behavior and limitations of ceramics, including lack of ductility and sensitivity to stress concentrations. Attachment to metal components with different thermal expansion properties requires further development. Finally, the ceramic components must demonstrate the ability to withstand many start/stop, acceleration and deceleration cycles characteristic of automotive operation.

3.4.2.2.3 Water Injection

By injecting water or other suitable fluids into the engine airflow, the thermodynamic and physical properties of the working fluid are altered so that more work is derived per pound of flow. Injection could occur at the compressor or combustor inlets. Analysis has shown that water injection (water-to-air mass ratio of 0.06) at an ambient temperature of about 80°F, can increase engine power by 35 percent. However, this power increase is obtained at the cost of a reduction in compressor efficiency. At a water-to-air mass ratio of 0.06 the ratio of compressor efficiency with water injection to the compressor efficiency without water injection is 0.83, a substantial drop in compressor efficiency (Ref. 3-28). A positive result of the power-boost feature is that the engine size can be reduced, resulting in an improvement in part-load operating efficiency. NASA is supporting Chrysler's efforts to evaluate this concept under the EPA contract.

3.5 PROJECTED STATUS

3.5.1 Potential for National or Regional Transportation

EPA studies conducted by engine manufacturers indicate that the gas turbine has good potential for use as a future automobile engine. However, automobile manufacturers have to be convinced of the economic advantages of the gas turbine over the spark ignition engine before it can become a strong candidate for future automobiles. In addition, the following technical problems have to be solved.

- The efficiency of the engine has to be increased to improve fuel economy. Development of a ceramic turbine will aid in solving this problem by allowing a significant increase in the turbine-inlet temperature.
- Emissions have to be lowered below the original 1976 Federal emission standards. Continued development of advanced combustors is the major approach taken to solve this problem.
- Additional improvement and development of ceramic materials is required to enhance their suitability for regenerator, turbine, and other high-temperature, high-stress applications in the gas turbine engine.
- Techniques have to be developed for mass production of cast, high-temperature, super-alloy components.
- Methods have to be developed to verify low production costs for control systems and regenerators.

It is anticipated that a period of from 7 to 11 years will be required before gas turbines can be mass produced for automobiles. A follow-on to the current EPA program with Chrysler can be conducted to develop a low-production version of an automobile gas turbine engine. As additional research and development results become available, this engine can be modified if production processes permit.

3.5.2 Impact on Energy Requirements and Air Quality

The Chrysler baseline engine program is directed toward proving that significant reductions in exhaust emissions and major improvements in fuel economy are practical with gas turbine engines. This new engine could then be a major contributor to the reduction in air pollution.

With the development of high-temperature turbine materials and operation at 2,500° F, the efficiency of the gas turbine cycle should increase to the point where a 20 percent fuel economy improvement over current gas turbine road performance can be anticipated. This would allow gas turbine-powered automobiles to be better than currently projected for piston-engine cars controlled to the original 1976 Federal emission standards.

High-octane fuels are not necessary for the gas turbine; thus, the issue of fuel lead additives as a potential air pollutant and toxilogical hazard would be eliminated as a concern with the use of these engines. Indeed, vehicular gas turbines can use a variety of fuels, ranging from natural gas to diesel and heavier oils, thereby providing the potential for making maximum use of available fuel resources.

4. RANKINE CYCLE ENGINES

4.1. INTRODUCTION

4.1.1 General Description

The Rankine cycle engine is an external combustion engine in which high pressure steam or some other working fluid is expanded in a reciprocating (piston-type) or turbine-type device to produce work. Figure 4-1 shows the components of a typical Rankine cycle engine in schematic form. The feed pump draws liquid at a low pressure from the reservoir and forces it under high pressure into the vapor generator where it is converted to superheated vapor by heat from the combustion gases. The hot, high pressure vapor then flows into the expander where it gives up energy as it expands to a lower pressure and temperature. Exhaust vapor from the expander enters the condenser where it is converted back to a liquid by rejecting heat to the surroundings. The resulting low pressure liquid is then returned to the reservoir or sump to complete the cycle. The major components shown (feed pump, vapor generator, expander, condenser) are required for any Rankine cycle system. Specific systems will show variations in such features as type of drive for the auxiliaries, additional heat transfer paths from one part of the cycle to another, and the type and location of liquid reservoir and boost pump.

4.1.2 Historical Background

Rankine cycle steam cars held a prominent place in the early days of the automobile. Steam cars were sold commercially in the late 1800s and early 1900s with familiar names such as Stanley Steamer, Locomobile, White and Doble. These early steam cars had a simple pressurized boiler containing a large quantity of water which required about a 30 minute cold startup, and posed a continual safety problem. They were noncondensing, dumping exhaust steam overboard. Consequently, frequent stops were required for makeup water. In spite of these disadvantages some of these cars, such as the

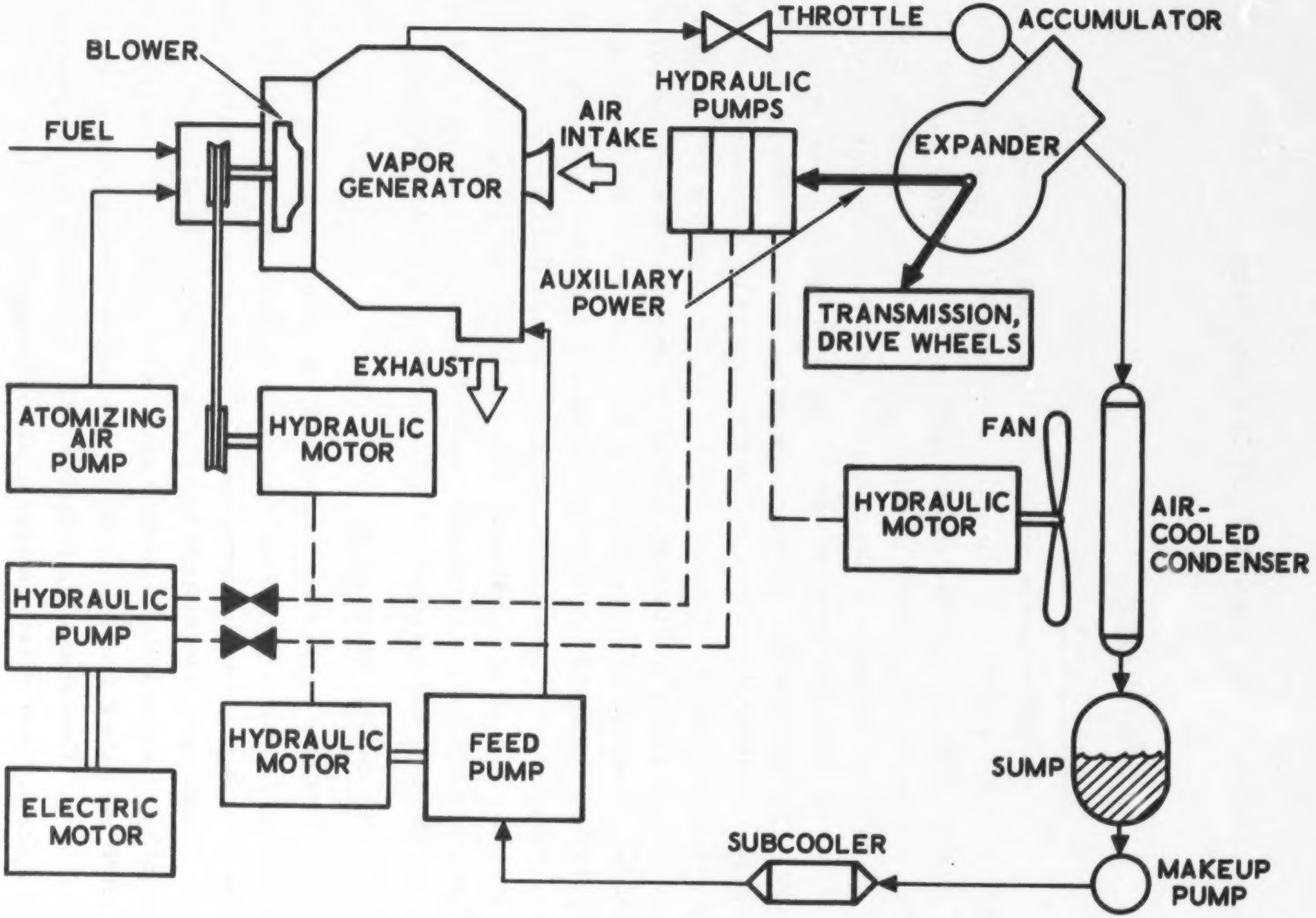

BLOWER
FUEL
VAPOR GENERATOR
AIR INTAKE
HYDRAULIC PUMPS
THROTTLE
ACCUMULATOR
EXPANDER
AUXILIARY POWER
TRANSMISSION, DRIVE WHEELS
EXHAUST
ATOMIZING AIR PUMP
HYDRAULIC MOTOR
FAN
HYDRAULIC MOTOR
AIR-COOLED CONDENSER
HYDRAULIC PUMP
HYDRAULIC MOTOR
FEED PUMP
ELECTRIC MOTOR
SUMP
SUBCOOLER
MAKEUP PUMP

Figure 4-1. Typical Rankine Cycle System, Schematic

Stanley Steamer, were quiet-running cars of quite satisfactory performance. Various technical advances were made with the years, primarily the monotube boiler by Doble, which effectively solved the safety and startup problem. Condensing radiators were in use from about 1915 on, resulting in true closed Rankine cycle operation. By this time, however, the internal combustion engine had entered its era of low-cost mass production. In this respect the steam car did not compete, disappearing from the automotive scene in the very early thirties.

Interest in the steam car occurred sporadically during the fifties and early sixties, but it was not until the latter sixties, with the impact of impending air pollution control legislation becoming apparent, that the Rankine cycle automobile system began to receive serious scientific attention. The impetus behind this renewed interest lay in the clean burning characteristics of the external combustor of this system, which requires neither post-engine exhaust cleanup devices, nor compromises with an efficient burning process.

A number of engineering efforts were undertaken at about this time by widely diverse organizations. This development received impetus and cohesion with the initiation in 1970 of the EPA Advanced Automotive Power Systems (AAPS) Program. The purpose of this program was, first, to identify the key problem areas and development tasks, and then to proceed with experimental and analytical programs which would result in an accurate appraisal of the overall feasibility of the Rankine cycle engine as a practical, mass produced power plant. Table 4-1 is a summary of the more significant worldwide Rankine cycle efforts which are either completed or now under way.

4.2 POWER PLANT DESCRIPTION

A Rankine cycle engine is one in which the heat generated by combustion is transferred to a second fluid, which then does work on mechanical components to provide engine power. A major reason for current interest in Rankine cycle engines is based on this external (to the working fluid) combustion. Since the combustor is required to provide heat only, combustion pressures can be atmospheric; maximum combustion temperatures can be

Table 4-1. Summary of Rankine Cycle Engine/Vehicle Development Programs

Developer/Contractor	Vehicle	Rated (or Estimated) Horsepower	Type of Expander and Rated Speed in rpm	Working Fluid	Status of Program
EPA AAPS Program					Preprototype engine development program completed in December 1973. One contractor (SES) selected to proceed with prototype engine development, followed by installation in vehicle.
Aerojet Liquid Rocket Co.	6-Passenger car[a]	150, Expander gross	Turbine; 32,000	AEF-78 (Proprietary)	
Lear Motors Corp.	5-Passenger car[a]	123, Expander gross	Turbine; 65,000	Water	
Scientific Energy Systems	6-Passenger car[a]	158, Expander gross 138, Transmission in	Piston; 2,500	Water	
Thermo Electron Corp.	6-Passenger car[a]	146, Expander gross	Piston; 1,800	Fluorinol - 85	
DOT/California Steam Bus Program					Program completed September 1972. Consisted of demonstration use in public service and vehicle emissions testing.
Brobeck & Associates	51-Passenger bus	240, Gross bhp 200, Net bhp	Piston; 2,100	Water	
Lear Motors Corp.	51-Passenger bus	240, Gross bhp 180, Net bhp	Turbine; 65,000	Water	
Steam Power Systems Co.	51-Passenger bus	275, Gross bhp 224, Net bhp	Piston; 1,850	Water	
DOT/Dallas Program					Program completed. Consisted of service in transit system and vehicle emissions testing.
Sundstrand Corp.	25-Passenger bus	120, Expander gross 90, Gear box out	Turbine; 35,000	CP-25; a single chemical compound comprised of methyl benzene, toluene, and toluall.	
California Clean Car Project					Testing and vehicle installation in progress. Target date for vehicle delivery is May 15, 1974.
Aerojet Liquid Rocket Co.	4-Passenger car[a]	60, Expander gross	Turbine; 60,000	Water	
Steam Power Systems Co.	4-Passenger car[a]	65, Expander gross	Piston; 2,400	Water	
Other Domestic Programs					
Carter Enterprises, Inc.	4-Passenger car (Volkswagen)	70	Piston; 5,000	Water	Vehicle first driven March 1972. Performance measurements being made.
Kinetics, Inc.	4-Passenger car	—	Gerotor type	Refrigerant 113	No published emissions or fuel economy data.
Williams Engine Co.	4-Passenger car	—	Piston	Water	
Foreign Programs					
Pritchard Steam Power Proprietary, Ltd.	6-Passenger car	—	Piston	Water	
SAAB SCANIA	Passenger car[a]	160	Piston	Water	Component development. Recently initiated.
Politecnico Milano	Bus[a]	100	Turbine	Organic	

[a]Simulated or installation design target. bhp = brake horsepower DOT = U.S. Department of Transportation

relatively low without strongly affecting cycle efficiency, and the flow rate of the combustion gases can be relatively steady (noncyclic). Under these inherent conditions, there is good evidence that efficient engine systems can be developed which emit very low air pollution.

4.2.1 General Power Plant Configuration

The basic ideal Rankine thermodynamic cycle for water is shown schematically in Figure 4-2. Starting at point A, the subcooled liquid is pumped up to boiler pressure, point B, with a feed pump. Heat from the combustor is then added at constant pressure in the boiler. The three main thermal portions of the boiler are shown as the heating section (line B - C), boiling (line C - D), and superheating (line D - E). The steam leaves the boiler at pressure and temperature conditions, E, and enters the expander. Work is extracted in the expander by rotary or reciprocating mechanical devices, and the steam conditions drop to those at point F which for purposes of demonstration is shown as still lying in the superheat region. In some systems, particularly those using organic working fluids (see Section 4.2.2.6), some of this remaining heat is transferred to the working fluid (from process line F - G to process line B - C) in a regenerator. The water then enters a condenser, at point G, where sufficient heat is removed and rejected to atmosphere to condense all of the remaining steam at constant temperature (line G - H) and to accomplish some subcooling of the liquid (line H - A). Very early steam engines often simply dumped the steam from the expander (at point F) to atomsphere in a so-called open cycle and drew fresh water from a stored supply at condition A (letting the environment close the cycle). All of the modern practical systems for automotive vehicles close the cycle through the condenser, as shown in Figure 4-2, to conserve the working fluid and thereby maintain acceptable vehicle travel range.

Basically, the work derived from the cycle of Figure 4-2 is the difference between the areas under the curves A - E and F - A. Efforts

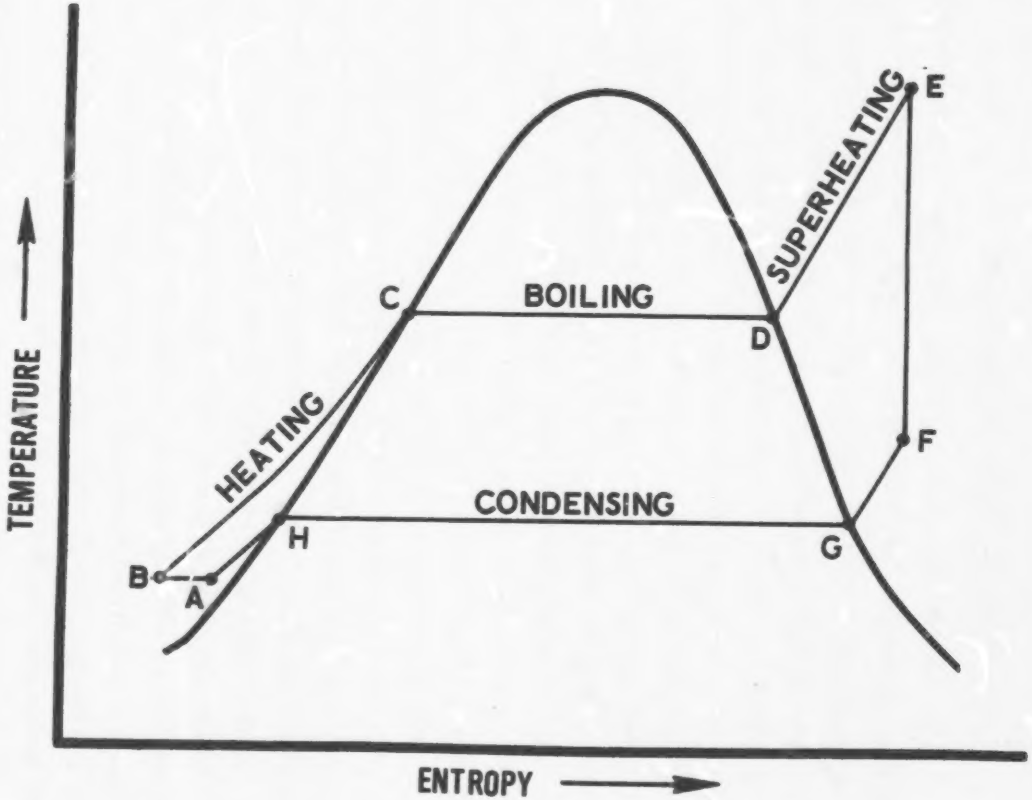

TEMPERATURE
ENTROPY
SUPERHEATING
E
BOILING
C
D
HEATING
F
CONDENSING
H
G
B
A

Figure 4-2. Typical Ideal Rankine Thermodynamic Cycle with Water as a Working Fluid, Schematic

to maximize the efficiency of the cycle, then, are primarily concerned with practical methods to: (1) raise curve A - E (high boiling and superheat temperatures), (2) lower curve F - A (low condenser temperature) and (3) make the actual thermodynamic processes in the expander approach the ideal isentropic process described by the vertical line E - F. Secondary efficiency gains can be obtained in a given system by transferring internally to other parts of the cycle any excess heat (which might otherwise be rejected to the atmosphere) from the condenser or the combustor exhaust. Some systems also use heat in the combustor exhaust to preheat the incoming combustion air or use available energy in the expander exhaust to drive parasitic components.

4.2.2 Component Design Features

Each of the components making up the Rankine cycle engine system previously discussed perform certain functions and present individual design problems. A brief discussion of each of these components, including some of the more significant and general design approaches, is contained in this section. For purposes of this discussion, the combustor and vapor generator will be considered as one component. In most cases they are, physically, one integrated component.

4.2.2.1 Feed Pump

The Rankine cycle requires relatively little work for the pumping process because this involves the liquid phase only. Feed pump work is not negligible, however, when vapor generator pressures are high. The design problem is to maximize the pumping efficiency and reduce the parasitic work required to drive the pump. This problem is compounded by the wide range of fluid flow rate required and available input shaft speed resulting from combined requirements of vehicle power demand and engine speed. The pump may be required to deliver the same full flow rate against high vapor generator pressures at both low ("lugging") and high engine speeds. In order to avoid fouling of the vapor generator heat transfer surfaces it is desirable to lubricate the pump with working fluid only.

Current designs involve a variety of piston pumps, including constant-speed, variable-stroke and variable-speed, fixed stroke designs as well as other multiple-stage pumps, sliding-vane pumps, and gear pumps. The parasitic horsepower (hp) requirements run from about 2 to as much as 8 percent of the gross engine hp. Special studies on feed pumps were conducted by Lear Motors and Chandler Evans Control Systems (Ref. 4-4).

4.2.2.2 Combustor/Vapor Generator

Currently three main air pollutants emitted from automotive combustion systems are of concern: oxides of nitrogen (NO_x), carbon monoxide (CO), and unburned hydrocarbons (HC). It is generally agreed that NO_x emissions can be minimized by avoiding long periods of combustion at high temperature in the presence of excess air. Complete oxidation of CO and HC, however, somewhat requires the opposite conditions. In engine systems wherein combustion gases also act as the working fluid, maximum cycle efficiency depends on achieving maximum combustion gas temperatures. By contrast, peak combustion gas temperatures in a Rankine cycle engine can be kept at a much lower level, dictated only by the requirements of efficient combustion and of providing adequate temperature gradient to the vapor generator. This low combustion temperature can easily be provided over a wide range of fuel air mixture ratios at atmospheric pressures and with appreciable dilution of the reactants with chemically inert gases (such as recirculated exhaust gases), if desired, without strongly affecting cycle efficiency.

The main design problem with the combustor is that of supplying heat efficiently to the boiler without emitting large quantities of undesirable air pollutants into the atmosphere. Current combustor designs are of both the direct-injection type and the prevaporization, premixing type with flame holders. Most use some degree of exhaust gas recirculation for further reduction of NO_x emissions.

As with any thermal cycle, there is a loss of efficiency in the energy rejected to the atmosphere from the hot exhaust gases. The temperature of the exhaust gases leaving the vapor-generator tube bundle must, of course, be higher than the inlet temperature of the working fluid into the tube bundle.

The latter temperature is essentially that of the condenser discharge (assuming no regenerator). Allowing for reasonable thermal gradients, this fixes the lower limit of the exhaust gases, as they exit from the vapor generator tube bundle, in the range 200 to 250°F. Values in this range are attained for actual systems at the lower heat-release rates. The exhaust temperature must rise for higher heat-release rates, however, and can become 600°F or more at maximum burn rate. Some of this heat may be retained by transferring it to the blower inlet air, and this is done in some systems. This produces a gain in overall thermal efficiency at the cost of increased blower-hp requirement and possibly some combustor design modifications required to accept the higher inlet air temperature. Attainment of maximum overall efficiency in a Rankine cycle system is dependent on the careful application of such tradeoffs for each particular system.

As discussed earlier, high efficiency steam Rankine cycle engines require a high pressure and high temperature working fluid in the boiler. The steam boiler design problem, then, is not only one of size and weight but also one of safety. The general approach for safety taken by most modern designers of automotive steam engines involves variations of the "mono-tube" concept which incorporates all of the boiler heat transfer surface in a single (or at most a few) coils of small diameter tubing. Should a tube rupture, only a small amount of energy in the vicinity of the rupture can be involved while the remainder can only bleed down slowly through the long tubing. These designs are considered explosion proof. Tube ruptures have occurred in development engines, but none have represented more than an inconvenience.

Special combustor/vapor generator studies have been conducted by Solar, Battelle, and General Electric (Refs. 4-4 and 4-5).

4.2.2.3 Expander

The two basic classes of expanders receiving most attention to date are the reciprocating (piston) type and the turbine wheel. The wide variety of past and current expander designs result from efforts to make the expansion (work producing) step approach as closely as possible to the ideal isentropic process indicated in Figure 4-2 as the vertical line E-F.

The major losses in reciprocating expanders are:

a. Mechanical friction of moving parts
b. Heat loss by conduction and radiation
c. Vapor leakage past valves and the piston
d. Incomplete expansion of the vapor
e. Fluid friction or throttling of the vapor
f. Initial condensation of hot vapor in contact with relatively cold surfaces in the cylinder.

The single most significant potential loss is the last of those described above, (f). The most common cause of cold surfaces is the cooling induced by the vapor itself after it has been expanded to low pressures and temperatures. One way to minimize this surface cooling is to keep the cool, expanded vapor away from the hot intake end of the cylinder. In the "uniflow" design concept this is accomplished by arranging the intake and exhaust ports of a cylinder at opposite ends. Engines of the uniflow type can deliver 73 to 83 percent of the ideal cycle efficiency, while those exhausing the vapor at the same end as the intake may deliver only 56 to 65 percent. Uniflow expanders tend to be larger in size, however, and may have greater mechanical friction losses.

Losses resulting from partial expansion of the vapor in a cylinder become important in torque considerations. If the full expansion of high pressure fluid from the vapor generator is taken in a single cylinder, the average force on the piston will vary widely during the stroke and the average torque will be low. Horsepower delivered to the wheels can be maintained by operating the engine at high rpm, but this creates a need for a variable-speed transmission. If the intake valves are left open longer, and full vapor-generator pressure is applied to the piston over a larger fraction of the power stroke, better torque characteristics are obtained. This "variable cutoff" concept may entirely eliminate the need for a transmission. But the vapor in the cylinder after a late cutoff (closing) of the intake valve cannot fully expand and a significant partial expansion efficiency loss occurs under certain conditions. (At part load the efficiency can be very high). With knowledge of the load cycle, an optimum combination of variable cutoff and transmission can be selected. Most current automotive reciprocating Rankine engines make use of some degree of variable cutoff.

Another way to improve torque characteristics and to minimize condensation losses is the "compound expansion" concept. Here the vapor is partially expanded in a small cylinder, at high mean pressures, then transferred to one or more large cylinders for complete expansion. The latter cylinders have lower mean pressures but may provide the same torque because of the larger piston areas. In some design concepts the vapor is routed through the vapor generator between the high and low pressure cylinders for "reheat".

Mechanical friction represents a significant efficiency loss in reciprocating expanders because of the high piston speeds, pressures and temperatures and the relatively poor lubricating qualities of most working fluids. Most automotive reciprocating Rankine engines use some type of standard lubricating oil in the cylinders. It is generally necessary to separate any oil which gets into the working fluid before recycling to the vapor generator, because it may coat the high temperature tubes and reduce heat transfer efficiency. In some designs, extensive schemes are employed to introduce oil into the cylinders, then separate it out before the liquid returns to the vapor generator. It appears that currently available solutions to this problem are not fully developed and piston and cylinder wear continues to be a potential problem. At least one special study of this problem area has been conducted (Ref. 4-3).

Turbine-type expanders are free of the problems of the frictional wear and lubrication technique, and maximum turbine efficiencies are of the order of reciprocating expander efficiencies. Turbine efficiency is particularly speed sensitive, however. Hence, it is strongly affected by the widely varying speed and power demands in automotive applications. For acceptable levels of efficiency, the turbine must be operated at high speeds; integral speed-reduction gearing and a widely variable transmission are then required to maintain turbine speed close to that required for maximum efficiency regardless of vehicle wheel speeds. These necessarily high turbine speeds also lead to problems with turbine shaft vibration and sealing.

4.2.2.4 Regenerator

Regenerators are necessary in those engines using working fluids which exhaust from the expander in a highly superheated state, usually at temperatures well above the ambient sink temperature. For a reasonably efficient thermodynamic cycle and minimization of the heat rejection load on the condenser, as much of this superheat as possible must be transferred to the working fluid entering the vapor generator. The basic design problem in the regenerator is the tradeoff between size and weight of the regenerator and gains in cycle efficiency. Current regenerator designs for automotive Rankine cycle engines generally appear adequate, and there is not a great deal of effort being concentrated in this area.

4.2.2.5 Condenser

Whereas the regenerator's function is to retain heat within the cycle, the condenser's function is to reject heat to the atmosphere. Generally, condenser temperature and pressure remain constant as the vapor is condensed. At a given pressure level, the condenser (or a separate subcooler) cools the working fluid slightly below the condensing temperature in order to provide some cavitation margin at the inlet to the feed pump. As with the regenerator, cycle efficiency strongly depends on cooling the working fluid as closely as possible to the ambient sink temperature. This leads to large heat transfer-surface area, size, and weight. With limited frontal area in automotive applications, a tradeoff of size and weight versus cycle efficiency is required. One common method of improving the condenser efficiency is forced-air circulation through the condenser at low vehicle speeds; at high speeds, the ram air velocity may be sufficient. Fans are required, particularly for operation at high-power, low-speed conditions; these fans may require 10 percent of the engine horsepower at full load. Major advances in condenser design have been accomplished recently by the AiResearch Manufacturing Co., Division of the Garrett Corporation (Ref. 4-4).

4.2.2.6 Working Fluids

Steam is the most widely used and versatile Rankine cycle working fluid. At temperature levels required for adequate cycle efficiency, however, relatively high vapor generator pressures are required. Attempts to keep boiler pressures low by increasing the degree of superheat yield diminishing advantages in terms of cycle efficiency. Referring to Figure 4-2, and the previous discussion on the steam Rankine cycle, it is clear that maximum work would be obtained from the cycle, for given maximum/minimum temperatures were the area enclosed by the cycle in Figure 4-2 to be a rectangle. The figure also shows that superheating the steam (line D - E) raises the peak temperature but, because this process is represented by a small triangular-shaped area over only the right-hand portion of the cycle, it does not increase the area bounded by the cycle to any great extent.

Certain organic fluids have thermodynamic properties that tend to make the Rankine cycle, plotted as in Figure 4-2, look more "rectangular"; i.e., more work can be derived from an organic cycle between the same maximum/minimum temperature limits because saturation pressures at a given temperature are lower for the organic fluids. These fluids also normally have a positive slope to the saturated-vapor curve (rather than negative slope for steam) in the working region (line D - G) on a Temperature-Entropy plot. This means that little or no superheat is necessary to avoid condensation in the expander since expansion, rather than condensing the fluid as in steam systems, vaporizes and superheats the organic vapor.

One disadvantage of the organic fluids, however, also results from this same characteristic. After the desired pressure expansion (usually to one atmosphere pressure to avoid condenser leakage problems), the organic fluid is still at fairly high temperature and is still superheated. To reject this residual heat energy to atmosphere in the condenser would seriously degrade the cycle efficiency. Thus a regenerator is necessary to transfer excess heat back within the cycle.

A second disadvantage of organic fluids is the maximum temperature limitations imposed by thermal degradation. The rather complex molecules of these organic compounds begin to break down at rather low temperatures. Thus, while water cycles often operate continuously above 1,000°F, most organic fluids are limited to 600 to 700°F peak temperatures. Much greater care must be taken in vapor-generator design to avoid "hot spots" or high temperatures in the fluid film the walls of the tubes.

In addition to thermal stability, many organic working fluids have factors such as cost, flammability, and toxicity that do not apply to water. Candidate fluids do exist, however, in which at least the last two factors appear to be satisfactory. The corrosion factor may be greater or lesser for the organic fluid than for water. Most organic fluids, however, have a freezing point well below minimum ambient temperatures. The high freezing point of water is a definite disadvantage for a steam Rankine cycle engine in automotive applications. As a result of the multiple advantages and disadvantages, a wide variety of organic fluids have been investigated, some of which are considered proprietary by the developers. The two automotive organic Rankine systems currently under development use different working fluids. A recent study of optimum organic working fluids has been conducted by Monsanto Research Corporation, (Ref. 4-4).

4.2.2.7 Transmissions

It seems clear, from past and current automotive Rankine cycle applications, that all engine systems have a single set of operating conditions at which the engine delivers the best combination of high efficiency and low emissions. At other operating conditions the efficiency may fall off severly, the emissions may exceed required levels, or both. The off-optimum operating conditions are forced on the engine system by the widely varying combinations of wheel speed and power/torque demand of the vehicle. Therefore, development of an "infinitely" variable transmission would benefit all types of Rankine cycle engines. This type of transmission functions over a very large speed ratio (generally stepless) between the engine shaft and the wheels. The engine could then operate near its best speed at all times. This is particularly desirable in engines with turbine expanders since, as noted previously, turbine efficiency is a strong function of turbine speed. Reciprocating expanders are less sensitive to engine speed but could also benefit from this application.

The problems in developing new transmissions are with efficiency, size, and weight. Several advanced-design transmission studies have been conducted recently (Ref. 4-3).

4.2.3 Operating Characteristics

All Rankine cycle automotive engine systems currently under consideration are designed for minimal effect on normal driving habits associated with the passenger car. Thus, such vehicles would still be controlled by operator activation of a conventional accelerator and brake pedal. In principle, the startup time from cold start need be no longer than that of a controlled spark ignition engine. However, this area may still need some more development for vaporizing-type Rankine cycle combustors.

The handling qualities of the Rankine cycle vehicle in most maneuvers can be expected to be very similar to present-day conventional automobiles. There will probably be certain conditions, however, in which a reduced response will be noticeable, such as a sustained power demand following an extended low-power operation. This results from a certian minimum thermal inertia inherent in the transfer of heat from one fluid to another through a wall. Safety considerations dictate that current systems do not have a very large storage of high-pressure, high-temperature working fluid that can be drawn upon to meet such transient demands. Short-term demands, however, for sudden acceleration or for maneuvers such as passing other cars can be met adequately.

Internal controls to ensure proper component operation, to coordinate fluid flows, and to ensure safety are all automatic and can be complex in some designs. There are no fundamental problems in control system techniques for Rankine cycle vehicles; but the optimum arrangement for a low cost, reliable control system is a matter to be determined only with the aid of extensive testing on the engine dynamomenter stand and, subsequently, in the vehicle.

A key control parameter is the air fuel ratio. The degree to which the control system can maintain this ratio within prescribed limits (determined in the steady-state test phase) during all engine transients and over the full load range will determine the degree to which low emission levels will be realized in actual vehicle operation.

Air fuel ratio control is, of course, required for any variable-speed engine, internal or external, and the task is not necessarily more involved for the Rankine cycle engine as compared to the internal combustion engine. Rankine cycle engines, however, involve a third fluid, the working fluid, which must also be controlled in varying proportion to the rate of heat generation by combustion processes. The problem is compounded by the fact that power supplied to pump the various fluids is usually taken directly from the shaft; thus, power is supplied at shaft speeds which are not directly related to the pumping-power demand.

For example, in a Rankine cycle engine, if a signal for vehicle acceleration is received, the first control step might be to delay the closing of the inlet valves in a variable cutoff reciprocating expander or to increase the steam flow area in a variable admission rotary turbine expander, thus increasing the demand for steam. A type of "anticipatory" control system would simultaneously begin to activate those components generating steam. The combustion air blower, fuel pump, automizer drive (in some cases), and the working-fluid feed pump would all be called upon to increase flows and rate of combustion in a coordinated manner, despite the fact that the drive speed to these components from the engine has not yet begun to increase. As engine speed increases, the proper flow rates and proportioning of these flows would have to be controlled despite the increase in drive speed.

Control subloops in individual components may also be required. Many vapor-generator designs involve automatic feed-flow proportioning devices that permit bypass of some of the working-fluid flow around the vapor generator in order to control final vapor temperature. Some sort of fan control based on condenser outlet temperature may also be required.

4.3 PERFORMANCE CHARACTERISTICS

There are no inherent problems in building Rankine cycle engines in the automotive horsepower range. A number of Rankine cycle engines have been built and are being built to fit in the same space and to supply approximately the same power as existing internal combustion engines.

Although current preprototype automotive Rankine cycle engines being developed in the AAPS Program have not yet been installed and operated in vehicles, measurements of emissions and component performance at steady-state conditions combined with reasonably conservative calculations of system performance for a car operating over the Federal Driving Cycle (FDC), indicate that exhaust emissions of HC and CO will be about 10 percent of the original 1976 EPA standards; NO_x emissions will probably be on the order of one-half of 1976 emission standards. Care has to be taken in the starting-sequence design to avoid excessive HC emission during cold start. Emission measurements have not yet been performed under transient conditions using a fully automatic control system.

Fuel economy in many of the initial and preprototype designs was poor; however, current automotive preprototype Rankine cycle engines appear to be demonstrating fuel economy comparable to prototype internal combustion engines with 1975 emission controls (Ref. 4-7). Continued development work on Rankine cycle engines will be guided by the need to further increase fuel economy while maintaining low emissions.

Significant noise in Rankine cycle automotive engines appears to emanate from the condenser fans and from the high-speed turbine in rotary expander engines. The high-frequency turbine "whine" might, in some cases, represent a higher or more irritating noise level that in the internal combustion engine. Otherwise, these engines are considerably quieter than internal combustion engines. External acoustic measurements made during the California Steam Bus Project showed that noise levels from the Rankine cycle engine-powered bus were about 10 decibels lower than a diesel-powered bus (Ref. 4-11).

With the very low level of HC emissions from Rankine cycle engines, it is unlikely that odor will be a significant factor. In the California Steam Bus Project using diesel fuels, "only light odors, reminiscent of gas turbine or jet engine exhaust, were sometimes noted around the steam vehicles."

There is not yet sufficient experience with modern, compact Rankine cycle automotive engines to assess durability and maintainability. It remains to be demonstrated that condenser designs can function adequately in all environmental condtions. The large number of thin, closely spaced aluminum fins necessary for a compact high-performance design might be susceptible to foreign object damage. The need to reduce condenser fan parasitic power requires low-restriction, cooling-air inlet designs which may also fail to restrict entry of foreign objects. On the other hand, an Australian steam car, using an automotive radiator for a condenser, has been road tested extensively, and the manufacturer anticipates engine life will exceed 250,000 miles (Ref. 4-4).

Primary safety problems (other than those normally associated with hydrocarbon fuels) are related to the high pressures and temperatures of the working fluid; particularly, the potential flammability and toxicity of organic working fluids. Also, any combustion products resulting from burning of the working fluid by fire is a potential problem.

Uncontrolled escape of the high temperature working fluid would be an obvious danger, but this is relatively easy to prevent, due in part to the monotube vapor-generator design previously described. With careful selection, organic working fluids can also be safe. Both of the organic fluids being used in the two systems in the AAPS Program are considered nonflammable and nontoxic, as is the prime candidate selected in the Monsanto organic working fluid study. The use of water as the working fluid, of course, eliminates the toxicity problem.

4.4 CURRENT STATUS OF TECHNOLOGY

Some of the more significant developments relating to automotive applications of steam Rankine cycle engines are summarized in Ref. 4-11. While steam cars preceeded internal combustion engines (ICE), they were eventually driven out of existence by rapid development of low-cost, mass production techniques for the ICE. Rankine cycle engines are receiving renewed interest today because of the recent emphasis on low air-pollution emissions from mobile cources. No significant number of Rankine cycle automotive vehicles are in use today other than research vehicles. New technology in this direction is being fostered by the programs described below.

4.4.1 Current Research and Development

Two complete automotive Rankine cycle system development programs are currently under way: the EPA AAPS R/D Program and the California Clean Car Project. The DOT-sponsored California Steam Bus project was also recently completed. A number of component and working fluid studies are also current or have recently been completed. There are also a few steam-powered vehicles of interest under private development. The system programs will be discussed here first, followed by the component and private studies.

4.4.1.1 AAPS R/D Program

The AAPS R/D Program was initiated in 1970 to demonstrate alternative power systems to the spark-ignition, internal-combustion engine applicable to full-size passenger cars. Three Rankine system contractors were selected in 1971, and a fourth was added about a year and a half later. The four approaches represent the four combinations of water versus organic working fluids and reciprocating versus turbine expanders. The four system contractors are:

- Water Working Fluid:

    1. Reciprocating Expander. Scientific Energy Systems Corp. [(formerly Steam Engine Systems)(SES)], Watertown, Mass.

2. Turbine Expander. Lear Motors Corp. (LMC) Reno, Nevada

- Organic Working Fluid:

3. Reciprocating Expander. Thermo Electron Co. (TECO), Waltham, Mass.
4. Turbine Expander. Aerojet Liquid Rocket Co. (ALRC), Sacramento, Ca.

Development of a low-emission preprototype engine system by each contractor was the primary goal, followed by selection of a single prototype design for installation and testing in a vehicle. The contractors went through several iterations in component designs during the preprototype development phase. Proposed designs for prototype engines are nearing completion. Figure 4-3 shows a mockup of a typical installation in an engine compartment for a 6-passenger car. Figures 4-4 through 4-7 are photographs of the four different preprototype engines; Figures 4-8 through 4-10 indicate the manner of prototype installation of one of these systems in a conventional passenger car. Discussions of contractor progress with concepts, designs, design problems, and tests are contained in the AAPS Contractors Coordination Meeting Summary reports (Ref. 4-1 through 4-5, 4-7 and 4-8). Table 4-2 lists some of the significant preprototype power plant design features for each contractor. Proposed prototype designs are different in some details from those indicated in the table. Comments on the data presented in Table 4-2 in context with discussions of the previous sections follow.

Three of the four combustor designs involve vaporization of the fuel and mixing with air prior to entering the combustor, with the attendant need for flameholders. This technique is capable of producing very low steady-state emissions of all three pollutants but requires considerable design attention with respect to low-HC emissions during cold start. None of these combustor designs uses staged combustion, and three of the systems use varying degrees of EGR, for NO_x emission control.

Figure 4-3. 150-Horsepower Steam Engine (Scientific Energy Systems Corporation)

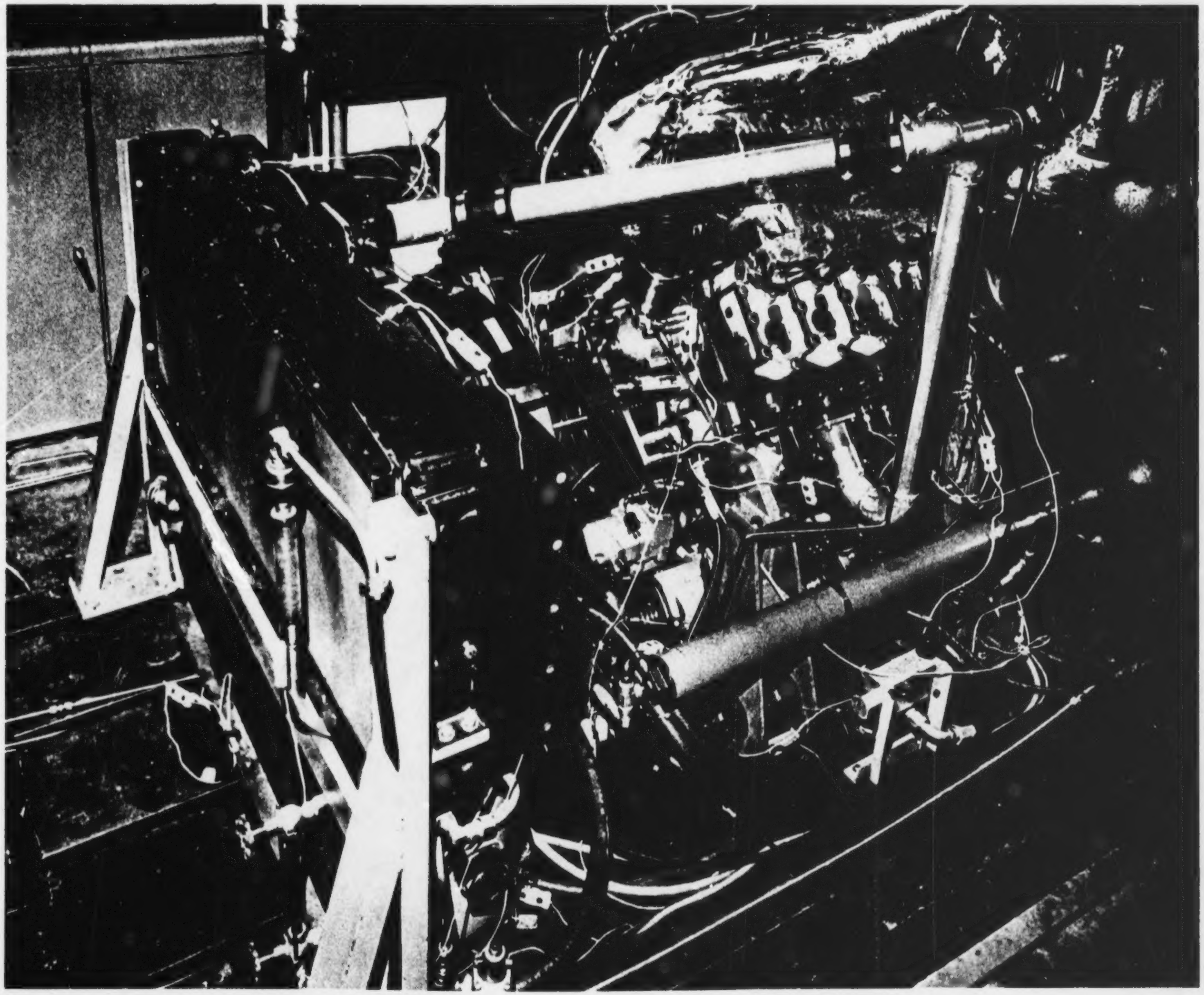


Figure 4-4. Steam Reciprocating Engine (Scientific Energy Systems Corporation)

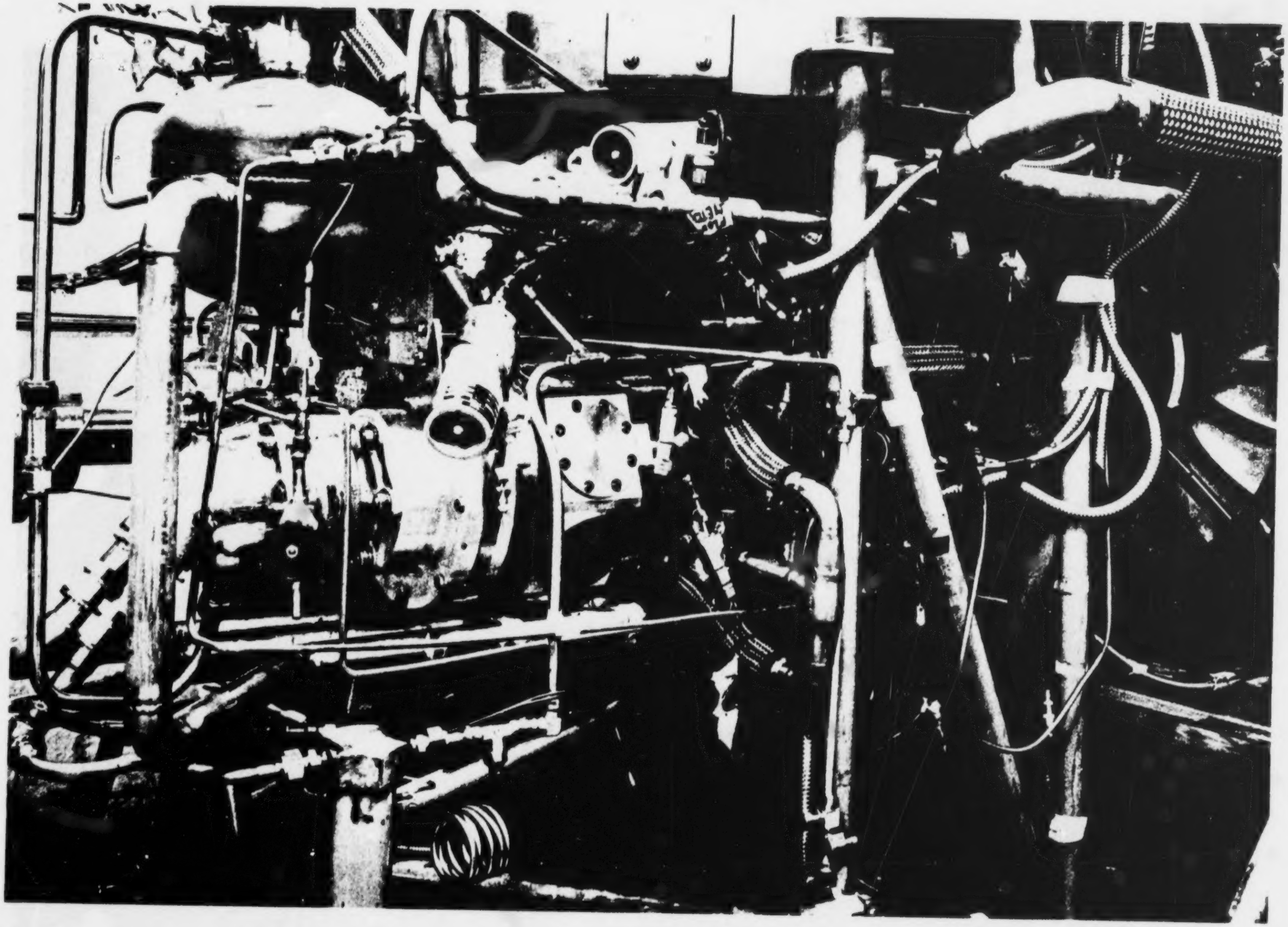


Figure 4-5. Organic Reciprocating Engine
(Thermo Electron Company)

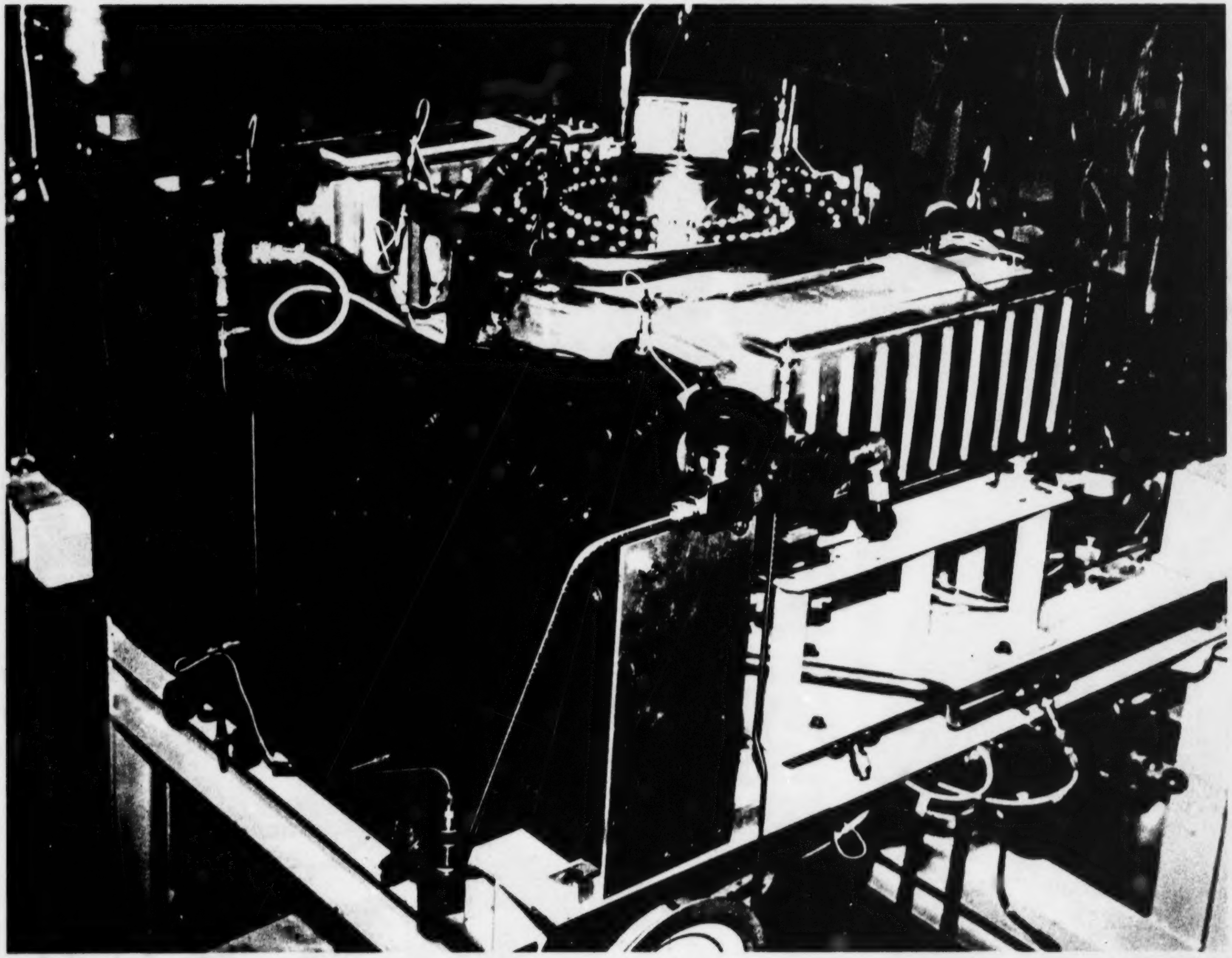


Figure 4-6. Organic Turbine Engine
(Aerojet Liquid Rocket Company)

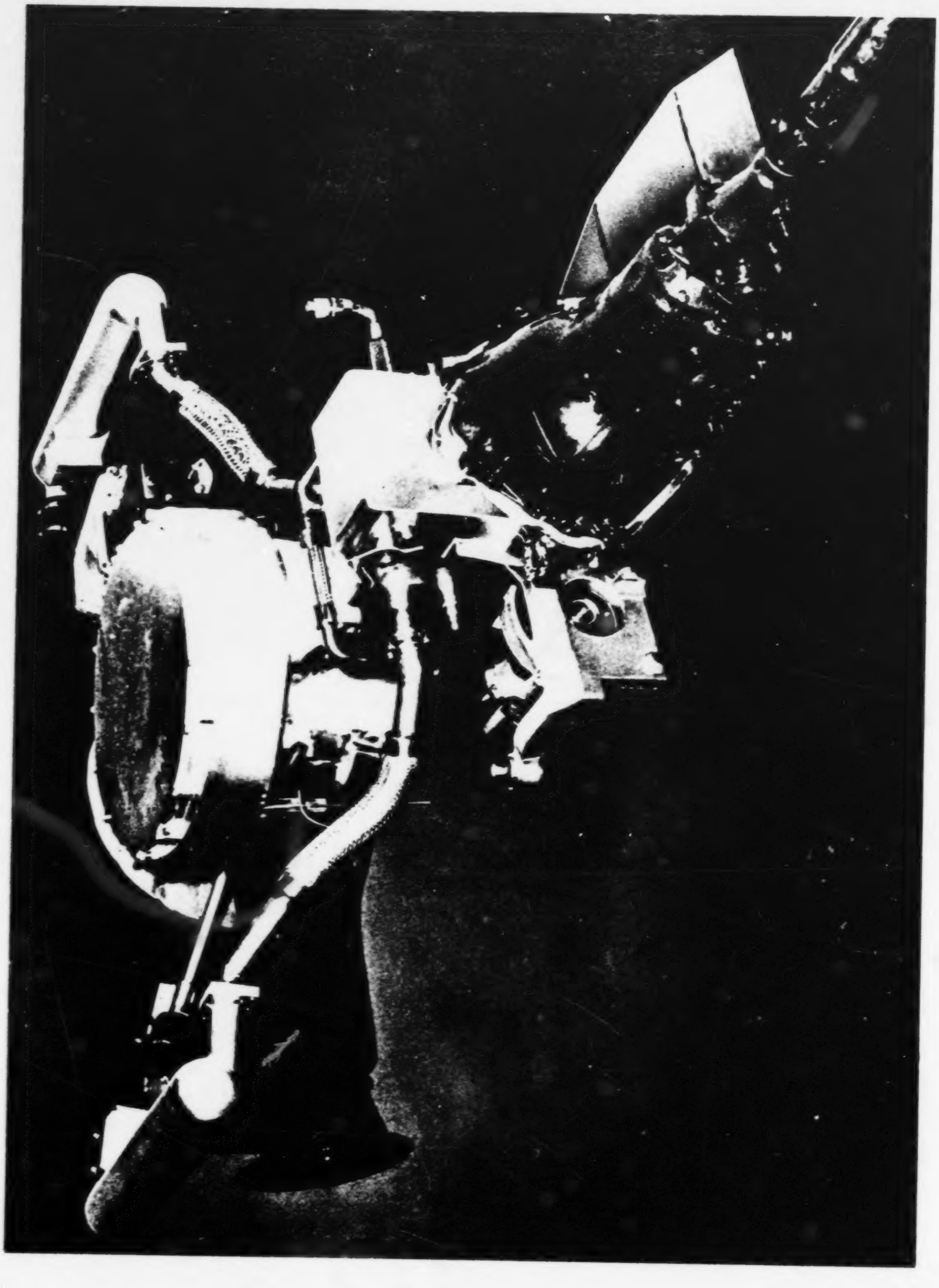


Figure 4-7. Steam Turbine Engine
(Lear Motors Corporation)

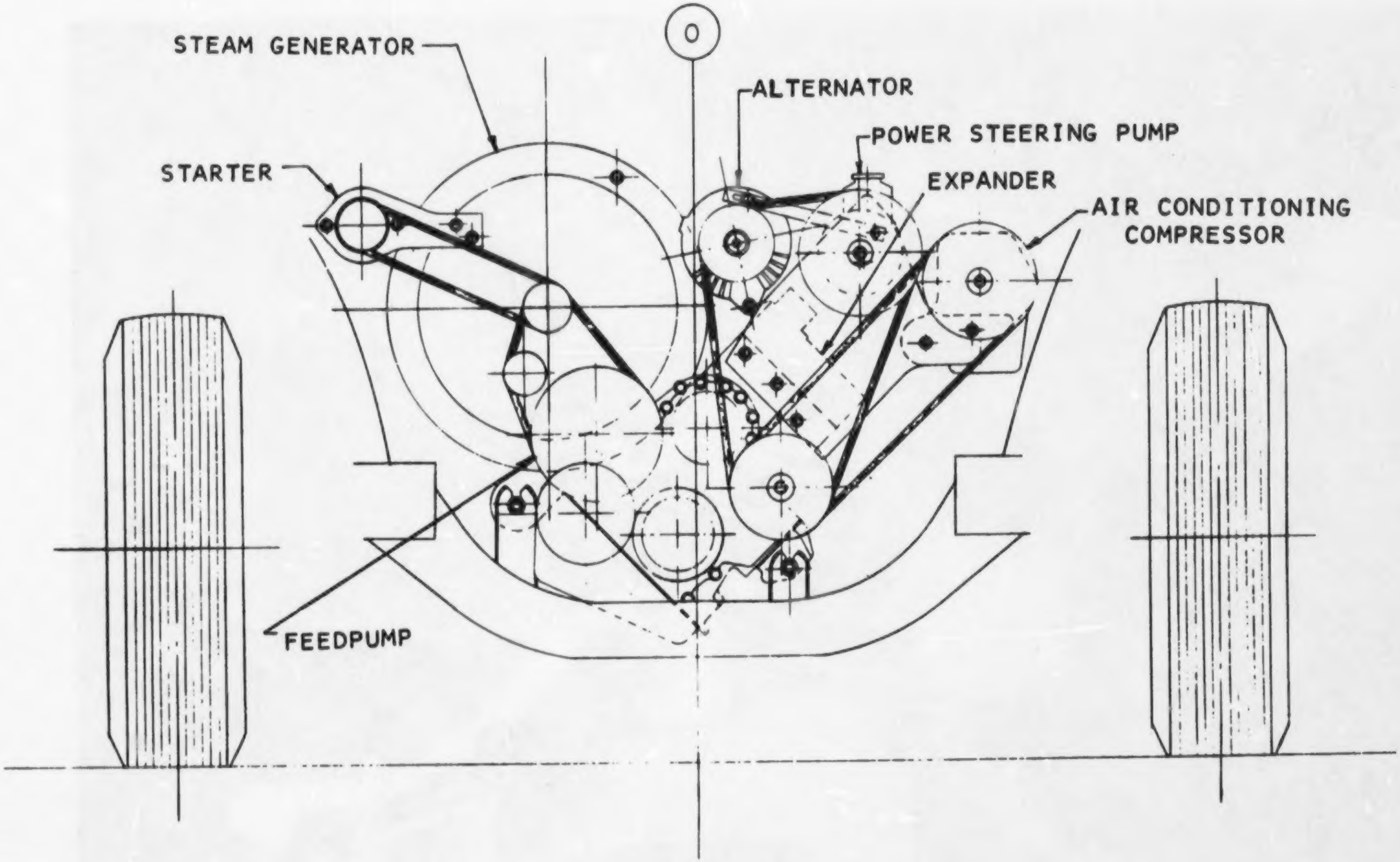

STEAM GENERATOR
0
ALTERNATOR
POWER STEERING PUMP
STARTER
EXPANDER
AIR CONDITIONING
COMPRESSOR
FEEDPUMP

Figure 4-8. Prototype Installation, Condenser Removed, Plymouth "C" Body
[(Front Interior View) (Scientific Energy Systems)]

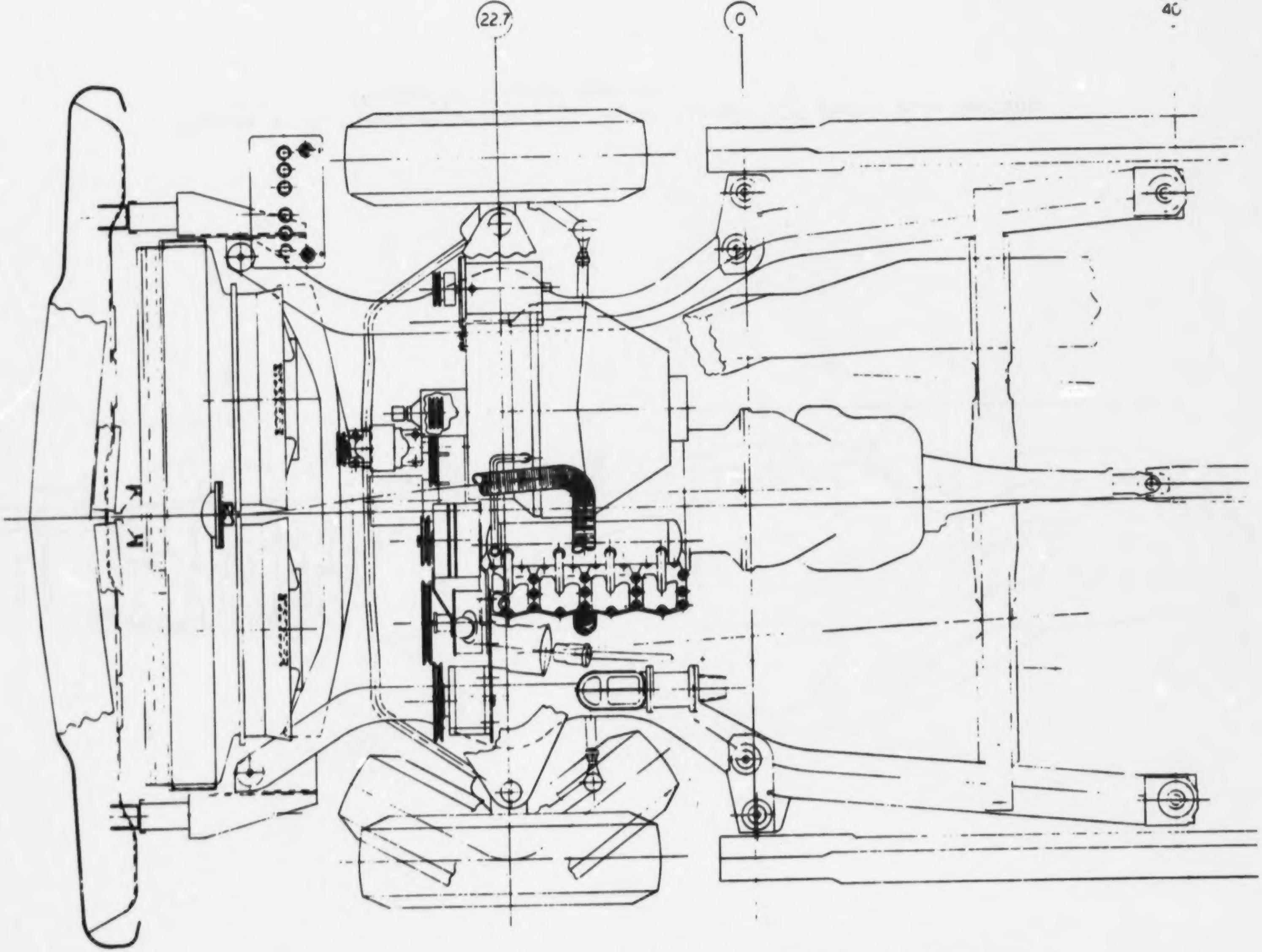


Figure 4-9. Prototype Installation, Plymouth "C" Body, Top Section (Scientific Energy Systems)

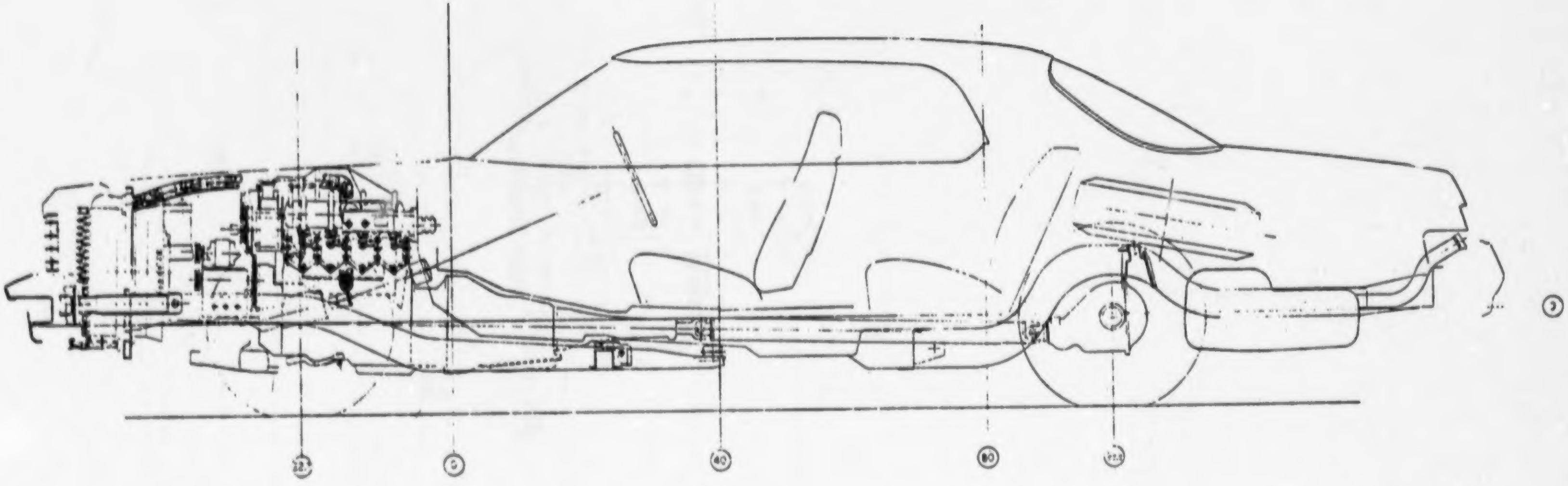


Figure 4-10. Prototype Installation, Plymouth "C" Body, Side Section
(Scientific Energy Systems)

Table 4-2. AAPS Rankine Cycle Engine Program Preprototype Power Plant Descriptions

Contractor	SES	LMC	TECO	ALRC
Working fluid	Water	Water	Fluorinol-85	AEF-78 (Supercritical)
Expander	4-in-line single-acting, trunk-piston uniflow	Single-stage supersonic impulse turbine	V-4 single-acting, trunk-piston uniflow	Single-stage supersonic impulse turbine
Combustor	Air-atomizer, pre-vaporizing, EGR	Air-atomizer, pre-vaporizing, pre-heater EGR	Direct-injection, rotary-atomizer, EGR	Pre-vaporizing pre-heater
Vapor Generator	Monotube	Monotube	Monotube	Supercritical, multi-tube
Regenerator	None	Steam-to-air	Vapor-to-liquid	Vapor-to-liquid
Condenser	AiResearch hydraulic drive	Harrison belt-drive	AiResearch belt-drive	AiResearch hydraulic-drive
Expander Control	Variable cutoff	Variable admission	Variable cutoff	Variable admission
Feedpump	3-in-line piston, variable delivery (direct drive)	Variable-displacement	Variable-displacement, radial-piston	Variable-displacement, vane

Vapor generators are predominantly of the monotube design.

Both organic working fluid systems use a regenerator; the water-based systems either use no regenerator (SES) or use excess heat in the expander exhaust to heat the incoming combustion air in a sort of hybrid regenerator/preheater (LMC).

All three of the initial contractors made use of the AiResearch plate and fin condensers supplied them by the government. The LMC condenser was manufactured by Harrison Radiator Division of General Motors.

In each case the major engine power control is exercised on the expander. The reciprocating expander designs use the variable cutoff concept with the resulting partial expansion losses and favorable torque characteristics discussed earlier. Turbine power control in both cases is obtained by varying the steam flow area (admission) to the turbine.

Preprototype feed pumps are generally of the variable-displacement type. There does not appear to be general agreement on the best feed pump design. This is also true of the two independent feed pump studies discussed later.

It appears too early in these current system programs to tabulate conclusive data for exhaust emissions or fuel economy. Much of the testing and effort to date has been in component development with low emissions as the primary design goal. Emissions are measured as concentrations of pollutants in the combustor exhaust during steady-state engine dynamometer tests. To properly extrapolate these data to emission levels expected from a full-engine system installed in a vehicle driven over the Federal Driving Cycle (FDC) requires accurate knowledge of: (1) engine system and power-train efficiencies over the full range of speed and power required in the FDC, (2) cold-starting, exhaust-emission characteristics and (3) the effects of engine system transients on emissions. To date, these data have not been fully established for any of the systems in the program. Based on the steady state data and what are believed to be reasonable analytical procedures, the conclusion is that the Federal Driving Cycle emissions will be quite low. These preliminary data are shown in Table 4-3. These computations may not adequately describe emissions during transient operation and HC emissions during cold start.

Table 4-3. Preliminary Emission Results on Federal Driving Cycle (gm/mi), Reference 4-12

	HC	CO	NO_x
Water Reciprocating	0.09	0.50	0.20
Water Turbine	0.21	0.70	0.38
Organic Reciprocating	0.02	0.19	0.25
Organic Turbine	0.16	1.01	0.13

NOTE: These values are based upon steady-state measurements and analytically converted to the 1972 Federal Test Procedure.

From the small amount of available fuel economy data, it appears that preprototype systems have demonstrated equivalent fuel economy to comparable spark ignition engines with emission controls. These data, again based on steady-state tests, are shown in Figure 4-11. This plot compares projected preprototype fuel economy data with preliminary data on 1975 spark-ignition, prototype vehicles. The data have been corrected to the same vehicle weight and test procedures.

Measured preprototype engine system weights tend to be several hundred pounds greater than the comparable spark-ignition engine. Comparable weight was not of prime concern in the preprototype design. Predicted weight reductions in prototype engines narrow this gap somewhat. If these predictions are borne out, the small addition to vehicle weight should not have a major effect on fuel economy and emissions.

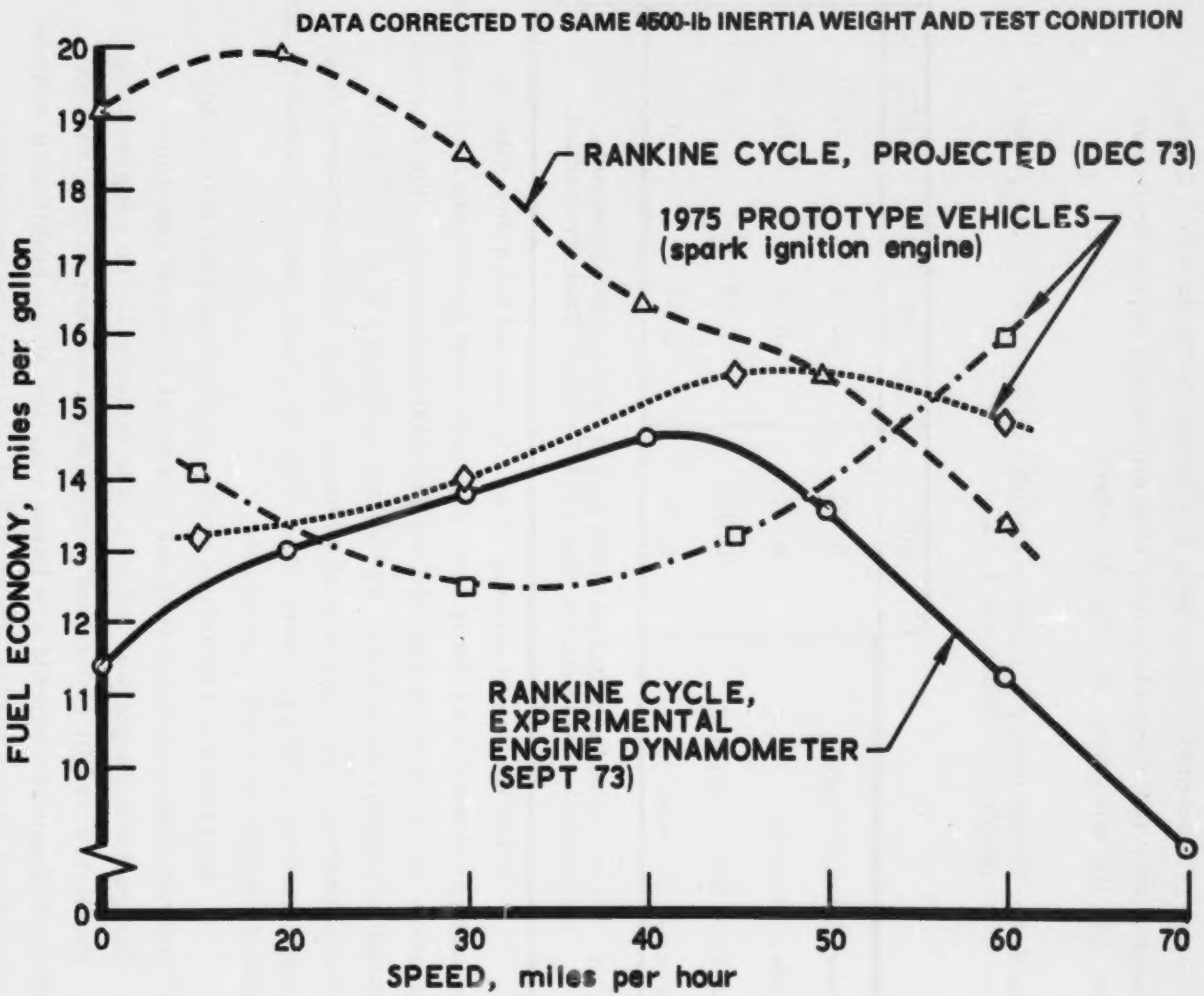


Figure 4-11. Current and Projected Fuel Economy for Rankine Cycle Engines (Ref. 4-12)

The large reduction in frontal area and volume achieved in the AiResearch condenser, discussed later, has contributed strongly to the success in automobile underhood packaging of each Rankine engine system under development. It seems clear, however, that all available space will be required. It remains to be demonstrated whether the high-density, thin, delicate plate and fin construction of these condensers is practical in the rugged usage associated with many typical automotive operations. Demonstrations must be carried out with a prototype system installed in a car and field tested.

4.4.1.2 California Clean Car Project

The California Clean Car Project [(CCCP) (Ref. 4-9)] was initiated in November, 1972 by the California State Assembly. This project differs from the AAPS Program primarily in that it addresses the compact car field. The AAPS Program considers a vehicle of about 4,300 pounds curb weight, while the CCCP requirements call for curb weight in the range 2,300 - 2,700 pounds. Aerojet Liquid Rocket Company (ALRC), Sacramento, California and Steam Power Systems (SPS), San Diego, California were the contractors selected. Initial evaluation of both cars is scheduled for summer, 1974. With little more than 18 months for each contractor to supply an operating vehicle, the project primarily addresses technological feasibility and low emissions with concern for fuel economy to be undertaken in a later program.

Both contractors are using water as the working fluid: ALRC uses a single-stage impulse turbine expander with gearbox as in its AAPS program engine, but in this case includes an infinitely variable hydrostatic transmission. The ALRC vehicle uses a monotube vapor generator procured from Scientific Energy Systems (SES) that is very similar to the SES vapor generator in the AAPS Program.

The SPS engine uses a four-cylinder, double-acting, compound-expansion expander with continuously variable cutoff. The steam generator is being developed by Solar Division of International Harvester. The combustor design is similar to that developed in a special combustor study for the AAPS Program (discussed later). Engine power and speed control is accomplished largely through variations in the expander steam cutoff, but a two-speed transmission is also used for achieving smooth torque and power changes to the rear wheels.

Both contractors are well into the component test stage and engine system assembly is about to begin. Preliminary system emissions and fuel economy data, again based on early steady-state component testing. appear to be in the same range as data obtained to date in the AAPS Program. Contractor estimates of exhaust emissions are all well below Federal 1976 standards, and fuel economy is predicted to be equivalent to the levels demonstrated by comparable spark-ignition engines. As yet no engine system or installed vehicle system tests have been conducted.

4.4.1.3 California Steam Bus Project

Although this report concentrates on alternative engines and fuels for automobiles, experience with related automotive projects (in this case buses) provides valuable data (Ref. 4-11).

The Steam Bus Project was initiated in June, 1970. Three contractors were selected:

a. William M. Brobeck & Assoc. (Brobeck), Berkeley, California in conjection with the Alameda Contra Costa Transit District (A-C) based in Oakland, California.

b. Lear Motors Corp. (LMC), Reno, Nevada, with the San Francisco Municipal Railway (MUNI).

c. Steam Power Systems (SPS), San Diego, California with the Southern California Rapid Transit District (SCRTD) of Los Angeles.

Funding was provided by the California State Assembly under a Federal research grant from the Urban Mass Transportation Agency. Overall project management and nontechnical evaluation was conducted by Scientific Analysis Corporation of San Francisco. International Research and Technology Corporation, Washington, D.C. provided technical management and evaluation services. Exhaust emissions were evaluated by the California Air Resources Board (ARB).

The project was completed in September, 1972 following an effort of about two years. All three of the buses were demonstrated in actual revenue service accumulating about 8,370 miles of road testing and service under steam power.

As in the California Clean Car Project, it was recognized at the outset that available time and funds were quite limited. As a result project objectives stressed the demonstration of practical, operating standard buses with low emissions and did not include any fuel economy goals. As a technical requirement, complete and design-inherent safety received the highest priority.

Standard 40-foot by 8.5-foot buses, originally configured to seat 51 passengers, were supplied to the contractors. As a time and cost-saving expedient, existing commercially available transmissions were employed, even though the torque converters were not well matched to either the reciprocating or turbine expander engines. For the same reasons, fixed-cutoff reciprocating expanders with steam throttling control were employed rather than the more difficult to develop, but more efficient, variable cutoff designs. Thus the gains in efficiency and torque characteristics obtainable from the optimized tradeoff of steam cutoff and transmission design were eliminated at the outset. The LMC system used partial-admission turbine control.

One distinct advantage of this project is that the three developmental Rankine cycle engines were installed in buses and operated over driving cycles as nearly identical as possible to those currently negotiated by optimized diesel engines. Thus the comparison between steam and diesel bus operating characteristics is direct, practical, and conservative. It is perhaps this direct comparison, rather than any absolute values for these steam engines, that is of greatest value.

Table 4-4 describes some of the salient design features of the three steam-bus engines developed. All systems used water as a working fluid largely because of the strong emphasis on safety and the

Table 4-4. California Steam Bus Project, Power Plant Descriptions

Contractor	Brobeck	LMC	SPS
Working Fluid	Water	Water	Water
Expander	3-cyl. double-acting, compound-expansion	Single-stage supersonic impulse turbine	6-cyl. double-acting, compound-expansion with reheat
Combustor	Air-atomizer, tangential-firing axial-flow	Air-atomizer/spinning cup atomizer radial-flow	Air-atomizer
Vapor Generator	Modified monotube	Monotube	Series/parallel-tube
Regenerator	None	Steam-to-air	None
Condenser	Remote hydraulic fan drive	Beltdriven fans	NA
Expander Control	NA	Variable admission	Steam-throttle
Transmission	Dana-spicer No. 184, 2-speed torque converter (direct above 29 mph)	Reduction gearing, 4-speed automatic (Allison HT-740)	GM-Allison super V (original bus transmission)

NA - Not available

inherent simplicity of water systems. Both contractors employing reciprocating expanders chose to use double-acting, compound-expansion designs ; and the SPS system uniquely included reheat of the steam between expansion stages. Some form of monotube steam-generator design, with its inherent safety, was used by all three contractors. The LMC engine was initially designed for an organic working fluid, requiring a regenerator. When the system was changed to water, the regenerator was retained in the system. Regenerators are not used in the other two systems.

Selected power plant specifications are shown in Table 4-5. Condenser frontal areas appear to be 3 to 6 times those of the AAPS Program

Table 4-5. California Steam Bus Project, Selected Power Plant Specifications (Ref. 4-11)

Contractor	Brobeck	LMC[(a)]	SPS[(a)]
Gross bhp	240	249	275
Rated net system bhp	200	180	224
Steam pressure, psi	1,000	1,000	1,000
Steam temperature, °F	850	1,000	750
Lowest BSFC, lb/net bhp-hr	0.985	1.13	1.18
Condenser frontal area, ft^2	34.5	19.4	32.2
Approximate weights, lb:			
Boiler with burners	920	890	850
Expander	965	110	1,250
Condensers with fans	750	420	800
Transmission	625	700(b)	600
Auxiliaries	491	392	800
Other	1,026	590	400
Total system, dry weight	4,777	3,102	4,700

(a) Derated from figures shown for use in bus system.

(b) With gearbox.

automotive condensers, although the bus-engine gross horsepower is less than twice that of the automotive engines. This reflects the improved AiResearch condenser designs in the AAPS Program. Engine system parasitic losses as fractions of gross horsepower appear to be in the same range as those in AAPS Program automotive engines. Only one of the bus-system designs (Brobeck) could be incorporated in available space without requiring a reduction in seating capacity. Both of the other buses had five less seats (nearly 10 percent reduction) than the original diesel-powered versions. However, the Brobeck system used the existing engine compartment only for the steam generator and its automatic controls; all other system components were located under the floor.

A comparison of average emissions of the three steam bus systems with those measured from four diesel powered buses is shown in Table 4-6.

Table 4-6. Comparison of Emissions from Steam- and Diesel-Powered Buses (Ref. 4-11).

	Emissions, g/bhp-hr			
	CO	HC	NO_x	$HC+NO_2$
Steam Buses	4.0	0.9	1.8	2.7
Diesel Buses	4.3	1.4	10.4	11.8
Calif. Heavy-Duty Engine Standards, 1975	25	-	-	5

These emissions data were obtained on the complete, installed engine systems, using a chasis dynamometer procedure over a standard driving cycle. On the average, all steam-bus emissions were less than those of the diesel buses. The average steam-bus NO_x emissions were only 17 percent of those from an average diesel bus and the HC emissions were 64 percent of the diesel. The average CO emissions from the two reciprocating expander-powered steam buses were 63 percent of the average for the diesel buses. (The LMC system tended to emit 2 to 3 times the CO of the other two Rankine systems). Thus, while the diesel engine is currently considered relatively "clean" in terms of CO and HC emissions, the steam systems tended to emit about two-thirds of these emissions. Also, the emissions shown above are in terms of grams of pollutant per brake horsepower-hour. The low steam bus emissions were obtained, then, in spite of the much larger fuel consumption of the steam engines for the same power supplied.

Fuel consumption of steam buses was 3 to 4 times higher than for the diesel buses. Under actual stop-and-go, bus-route driving conditions the steam buses obtained 0.52 to 1.1 mpg while the diesel buses delivered 2.3 to 3.7 mpg. Idle fuel consumption of the steam buses was more than 6 times that of the diesels. Some of the reasons for the low fuel economy of the systems demonstrated in this program were built into the program from the start, as discussed earlier regarding component design compromises. Other improvements could be expected in the normal course of extended development of this type of engine, with fuel economy a stated, high-priority goal. Certainly diesel engines, used as a standard of comparison in this project, are in a much more advanced state of development than are automotive steam engines.

4.4.1.4 Recent Component Technology Programs

Recognizing certain problem areas in the development of Rankine cycle automotive engines, special component studies have been conducted as part of the AAPS Program. Some of the more significant studies are concerned with problems of working fluid selection and optimum design of vapor-generator feedpumps, combustors, condensors, and transmissions. These are discussed briefly in the following pages.

4.4.1.4.1 Working Fluids

At least one special study of organic working fluid selection recently has been completed by Monsanto Research Corporation (Ref. 4-1 through 4-4). (To a lesser extent, organic working fluids have also been studied by a wide range of other contractors.) The objective was to identify superior working fluids for use in Rankine cycle automotive power systems using positive-displacement and turbine expanders. Selection criteria were performance, safety, economy, and acceptance (the latter category including all other important criteria). An initial goal of the study was that the cost of the initial fill plus replacement charges in a 5-year period should not exceed $100. Two final candidate fluids were identified:

		Mol %	Wt. %
a.	Pentafluorobenzene	60.0	57.5
	Hexafluorobenzene	40.0	42.5
b.	Water	65.0	26.4
	2-Methylpyridine	35.0	73.6

These two candidates were extensively characterized physically, chemically, thermodynamically, functionally, and economically.

Candidate (a) satisfies efficiency, temperature, pressure and density limitations, and regenerator size-limit criteria for both reciprocating and turbine-expander engines. It is considered fire-safe and unlikely to cause serious injury on accidental human exposure. It is stable up to at least 750°F. If allowed to escape to the atmosphere it degrades at a relatively slow rate. Synthetic lubricants are necessary with this fluid and several low-cost synthetics show promise. The fluid is compatible with low-cost, available construction materials.

Candidate (b) also satisfied all performance goals except the goal of 30-percent efficiency for an ideal cycle. It is flammable but less so than common fuels and is not likely to produce dangerous concentrations of toxic products in a fire. Accidental, external contact with the fluid is not likely to cause serious injury, although it is more toxic than candidate (b). It has an extremely disagreeable odor, which is viewed as a desirable characteristic in terms of positive indication of a system leak and avoidance of voluntary inhalation. It is stable up to 720°F. Metal corrosion poses a potential materials compatibility problem. The fluid, once released from the engine, would likely be rapidly decomposed by microorganisms and would not constitute an atmospheric pollutant. Compatible low-cost synthetic lubricants again appear necessary and promising.

Neither fluid appears likely to meet the cost goal, however. For candidate (a), cost to the owner of the Rankine cycle car for a 5-year supply of working fluid is estimated by Monsanto to be in the \$120 to \$200 range. The estimated cost of fluorinated benzenes is very sensitive to the assumed price of liquid fluorine. The 5-year cost for candidate (b) is estimated to be in the \$80 to \$150 range. Reasons for the wide range in the latter estimate were not stated in the references cited.

4.4.1.4.2 Feed Pumps

Each of the automotive Rankine cycle engine system contractors expended considerable effort on selection and development of working fluid feed pumps for their systems. Special studies were conducted by LMC and Chandler Evans Control Systems (CECO), Division of Colt Industries (Refs. 4-3 and 4-4). Pumps were designed to match requirements of the engine systems under development in the AAPS Program (both water and organic working fluids).

The separate pumps selected by LMC to satisfy all requirements of the engine systems were of the radial-piston type with variable capacity or variable speed and direct drive. Separation of the working fluid from the pump oil is provided. An externally-driven boost pump is necessary under certain operating conditions.

Vane-type pumps were selected by CECO as the best type for this application. The organic fluid pumps were two-stage, 6.25:1 flowrate turndown ratio, variable-speed drive. A fixed-displacement water pump employs a bypass. All pumps are lubricated by the working fluid. For a more near-term application, gear pumps are preferred because of the lesser development required.

All contractor designs met the requirements for overall efficiencies greater than 50 percent at 10 percent of maximum flow and 70 percent between 30 and 80 percent of maximum flow.

4.4.1.4.3 Combustors

Solar Division of International Harvester (Solar) initiated a combustor/vapor generator development program at the outset of the AAPS automotive Rankine cycle engine program (Refs 4-1 through 4-4). Combustor/vapor generator subsystems suitable for integration in the AAPS engine systems were to be constructed. Separate combustor studies were also conducted by Battelle-Columbus Laboratories (Refs 4-1 through 4-4) and, later in the AAPS Program, by the General Electric Co. (GE) (Refs. 4-3 through 4-5).

The Solar combustor involves a spinning-cup fuel atomizer injecting directly into the primary combustion space, with staged air admission. The air flow is split approximately equally between the primary and secondary combustion zones. The final vapor generator design was of the monotube-type with external fins on the preheater and superheater (dryer) tubes; an earlier parallel-tube design was subject to flow instabilities. Based on bench-level tests, all emissions and performance-level requirements were met except for NO_x at the very highest flow rates. No EGR was used.

Battelle investigated a wide variety of combustor designs involving primary zone cooling, staged-air admission (fuel-rich and fuel-lean primary) and EGR, all with an air atomizer producing "ultra-fine" fuel atomization. In general, the cooled, rich-primary configuration was predicted to just meet NO_x goals, and the addition of EGR was predicted to reduce NO_x well below these goals. Forecast HC and CO emissions were well within required limits.

The GE combustor program was initiated late in the AAPS Rankine cycle engine system program and was intended to investigate the feasibility of the use of a nonadiabatic, porous-plate combustor system for advanced automotive engines. Combustor firings and emissions measurements were obtained; satisfactory NO_x and CO emissions were obtained over a narrow range of loads. HC emissions were very low.

4.4.1.4.4 Condensers

A special study on compact condensers was initiated by AiResearch Manufacturing Co. Division of Garret Corp. in June, 1970 (Refs. 4-2 thru 4-4, also 4-6). The goal was to develop heat-transfer surfaces and design techniques that would lead to an automotive Rankine cycle engine-condenser design having frontal-area, volume, and fan-power requirements not appreciably greater than those of present automobile radiators. In the initial experimental-design study, a perforated-fin configuration was developed which was only 3 percent greater than the frontal area goal and 2 percent greater than the volume goal. The design is of brazed, all-aluminum construction, uses 22 fins/inch, and rejects about 1.5×10^6 Btu/hour. It can be mass produced at an estimated manufacturing cost of about $100 including materials, direct labor, and factory burden.

A follow-on development effort was conducted in which condenser/fan units were supplied to the three AAPS engine system contractors (using water and organic working fluids). These units have provided satisfactory service in the preprototype system test.

4.4.1.4.5 Transmissions

Sundstrand and Mechanical Technology, Incorporated, conducted two short studies on transmissions for advanced automotive engines (Ref. 4-3). Objectives of these studies were to compare technical and economic feasibility of transmission types; recommend a transmission for Rankine cycle automotive applications; and determine performance, cost, and characteristics of the transmission. All transmission candidates were of the infinitely variable type. Very early in the study Sundstrand rejected hydrostatic (poor efficiency), electric (poor efficiency, size, and weight), and hydrokinetic (no ratio control) systems. Hydromechanical transmissions were considered attractive for the near term, and traction type transmissions proved attractive with further development.

4.4.1.5 Other Efforts

There are a number of steam-powered vehicles in operation and under development. This section describes some of the more significant efforts under way both in the U.S. and abroad.

4.4.1.5.1 The Dallas Project

Under a grant from the Department of Transportation, Dallas Transit System subcontracted to Ling Temco Vought Aircraft Corporation, with a further subcontract to Sundstrand Corporation, to build and install an organic Rankine cycle engine in a small bus (Ref. 4-3). This bus underwent operational tests, emission measurements (before and after installation of the new engine), and limited operation on the transit system for evaluation. The specific objective of the program was to demonstrate operational feasibility of a turbine-driven, organic Rankine cycle engine in a 25-passenger bus.

The test vehicle was a 25-passenger, TC-25 transit coach originally powered by a 413 cu. inch Chrysler, V-8 gasoline engine with a Chrysler 3-speed automatic transmission.

The design criteria for the rankine cycle engine are:

- Supercritical cycle using CP-25 as a working fluid
- Same power-to-weight ratio as full-size city buses
- 700°F peak system temperature
- 235°F condenser temperature
- LPG fuel
- Nonprototypal baseline engine
- ASME Pressure Vessel Code where applicable
- Hermetic CP-25 loop.

The supercritical monotube heater uses a tangentially induced vortex combustor. The whole heater/combustor is 25 inches in diameter and 30 in. long including the combustor. Core weight is 150 pounds; total weight is 375 pounds. The single-stage, partial admissions, impulse turbine wheel is 6 inches in diameter, rated at 120 shp at 35,000 rpm (8,000 rpm idle) with a design point efficiency of 76.6 percent. The feedpump is a double-acting, variable-speed, fixed-displacement, axial-piston type, hydraulically driven. The regenerator is a finned tube cross-counterflow configuration with 12 aluminum fins/inch on copper alloy tubing. It is 27 inches long by 9.5 by 12 inches wide with a core weight of 120 pounds. The effectiveness is 0.90. The condenser is an air cooled, downflow configuration with aluminum fins on brazed-copper tubing. (Size: 4 ft. x 4 ft. x 5 in. thick). It is capable of 1,200,000 Btu per hour at 100°F air-inlet temperature. The 6.3-hp, axial-shrouded condenser fan is hydraulically driven (air flow 16,800 SCFM). Core and hotwell weight is 245 pounds.

4.4.1.5.2 Carter Enterprises, Incorporated

The Carter Steam Automobile engine has been under development for about 5 years. Approximately 3 years were spent on component development, which included four engine and five boiler configurations. After about 10 months on the test stand the current package was installed in the engine compartment of a standard Volkswagen automobile and driven for the first time on March 15, 1972. Engine description and characteristics are:

- Four cylinder, radial, 35 inch^3 with fixed 2 percent cutoff, 6 percent clearance and 14:1 expansion ratio
- 70 hp at 5,000 rpm (same rated speed as standard VW engine)
- 2,000 psi and 1,000°F rated steam conditions at throttle (unit has operated up to 1,100°F)
- Standard VW clutch and 4-speed transmission
- Total engine package weighs about 120 pounds more than the standard VW engine package
- About 30 to 40 seconds to drive off from a cold start
- An early fuel rate measurement showed 17.8 mpg at 55 to 60 mph. This was before finned tubes were incorporated in the boiler, and insulation was added to the cylinders and valves
- Start sequence is automatic with turn of ignition key; turndown ratio is 50:1; control is based on pressure drop across throttle valve

Status and plans for this development are:

- The prototype vehicle has been driven about 2,000 miles
- Performance and characteristics are being verified by third parties. Demonstrations will continue.
- Patent applications have been submitted for all appropriate parts, pieces, and systems of the engine.

4.4.1.5.3 Kinetics, Inc., Sarasota, Florida

This company has built an automobile utilizing an organic Rankine cycle engine system. Some funding for this work reportedly has been provided by the Datsun Company of Japan. The Kinetics system utilized a low-temperature working fluid (Refrigerant 113) and a Gerotor-type expander. The vehicle has no transmission. It is understood that three vehicles have been built and tested. Emissions or fuel economy data have not been published.

4.4.1.5.4 Williams Engine Company, Ambler, Pennsylvania

The Williams system employs a monotube steam generator and superheater with an oil-fired burner. Steam is delivered (through a throttle valve) to a four-cylinder reciprocating engine.

In its startup mode, patented auxiliary exhaust and regulating valves lower the compression pressure to permit maximum torque, with steam injected during five-eighths of the power stroke and expanded during the balance. However, in its "drive: and "economy" modes - equivalent to the higher ranges in a customary transmission - the injection of steam is cut off earlier in the cycle. Then, on the up stroke, a measured amount of residual steam is recompressed during which process its temperature becomes higher than that of the steam supply.

Early cutoff of the steam injection is accomplished by moving the valve camshafts so as to position different sets of cams under the valve lifters.

To complete the system, exhaust steam is passed through a water reheater and into a condenser of essentially full conversion into water. A pump then delivers water at high pressure to the steam generator as required.

Because of the variable cutoff feature, no transmission is required and the expander is coupled directly to the driveshaft. The cutoff selector replaces the conventional gear shifting mechanism, and the throttle valve functions in the same way as the customary foot accelerator. Appropriate sensors and automatic sensors and automatic controls regulate the generation of steam and safeguard the system.

In 1966 tests at Mobile Laboratories, the following emissions data were determined using the California 7-mode cycle test (Ref. 4-13):

	1968 National Standards	Williams Steamer 43 CID Engine (no anti-pollution devices)	
		As Measured	Corrected to Stoichiometric Air/Fuel Ratio
Hydrocarbons	275 ppm	20 ppm	33 ppm
Carbon Monoxide	1.5%	0.05%	0.08%
Oxides of Nitrogen	1,500 ppm	40 ppm	116 ppm

These measurements played a significant role in the renewal of interest in Rankine cycle automotive power plants, as they were significantly below levels of emission from contemporary internal combustion engines.

4.4.1.5.5 Other Developments

A number of other private groups are involved in some form of development leading toward a Rankine cycle powered vehicle. These are largely in the component development area, and some are briefly described here.

The Gibbs-Hosick Trust of Winston-Salem, N.C. has built a demonstration model of what they call an "Elliptocline" expander. The design is covered by U.S. Patent 3,370,511, and is essentially a variant of a barrel expander. They have a working model, but no test data to report.

Closed Cycle Systems, Maspeth, New York, is working on system incorporation of a steam turbine utilizing a pressed-steel disc for the blades.

D-Cycle Power Systems, Richmond, Virginia, is working on a cycle claimed to be novel, in which steam is recompressed to minimize heat addition. It is understood that they are building a system for test.

Horst Power Systems, Saratoga, California, has built a model of a rotary positive displacement expander. This has been run on steam in their laboratory. No data are available.

4.4.1.5.6 Foreign Programs

A number of programs to develop Rankine cycle engines are known to be under way outside of the U.S. Unfortunately, detailed information is not available at this time except for the activity described here.

Pritchard Steam Power Proprietary Ltd., Melbourne, Australia. Steam power systems have been in various stages of development in Australia over the last 20 years. An engine was developed and operated in a 5 ton truck in the early 1950s.

Extensive road tests were carried out on the Pritchard Engine in a 1963 Ford Falcon, in November, 1971, in Melbourne. Over 1,300 miles were run with an identical comparison car that was conventionally powered. Selected routes went through congested traffic areas as well as on freeways. Tests included acceleration, hill climbing, starting, noise, and emissions. The fuel was kerosene. Mileage over a varied 58.4-mile city course was 15.3 mpg. Measurements of NO_x showed 0.595 gms/mi. Performance characteristics were directly competitive with or better than the comparison car.

The system includes a monotube boiler, a uniflow piston expander with variable valve cutoff, and an exhaust-driven motor which drives the feedpump and the condenser fan. The condenser is an automotive radiator. The fuel pump and combustion air compressor are electrically driven. Roller bearings are used on the crankshaft and the large end of the connecting rods. Engine life is expected to exceed 250,000 miles. First cost is projected to be the same as the conventional engines; life time cost should be less.

SAAB-SCANIA. SAAB-SCANIA, Aerospace Division, Stockholm, Sweden, has a program which has been under way since 1968 (Ref. 4-10). Computer programs were initially developed to model engine characteristics, followed by an evolution of preliminary system designs and, lastly, by component development, which is now under way. Present hardware is being used as a development tool.

Auxiliaries and accessories are driven at constant speed by a single-cylinder auxiliary expander. Superheated steam at 1500 psi and 600 to 775°F is used. The vapor generator consists of a parallel arrangement of a large number of very small-diameter tubes in intimate thermal contact. These very closely spaced tubes permit laminar flow of the combustion products through the tube bundle. Vapor generator output temperature is controlled by a distribution valve between a single preheating section and two parallel boiling and superheating sections. The main expander is a 9-cylinder wobble plate device operating at 3000 rpm and directly connected to the differential. Power control and engine reversal are accomplished with a variable cutoff valve. The overall concept of the propulsion system eliminates the need for a transmission.

The condenser consists of two heat exchangers, one an air-cooled condenser sized for road-load conditions and a "condenser buffer" for transients. The condenser uses 80 fins/inch on both air and steam sides, with laminar flow on the air side. An "accordion" design is used to increase the effective frontal area.

SAAB-SCANIA indicates that the engine can develop 160 hp continuously and 250 peak hp for short periods. The total engine weight and fuel consumption are less than those of a conventional 100-hp engine, and space requirements are about the same. Emissions are considered "potentially low", and noise and vibration are considered "almost non-existent". The engine is still considered experimental.

Italian National Propulsion and Energetic Center. This center, located at the Politecnico in Milan, is performing research centered around a 100-hp bus engine. The engine will incorporate an organic working fluid, a high-efficiency multistage turbine expander, and an advanced regenerator. Emphasis in the design is to achieve as high efficiency as possible. This will, among other factors, reduce the waste heat and thus the condenser size. The use of regenerative braking is also being evaluated, with both mechanical and electrical methods being considered. A detailed user survey is presently in process. Although centered on the bus applications, the design can be scaled to passenger-car application.

4.5 PROJECTED STATUS

If it is assumed that the major determining factor in the emergence of a new type of automotive engine will be the need for simultaneous low pollution emissions and low fuel consumption, it then appears that the Rankine cycle engine represents a viable candidate. Current data indicate that emissions can be controlled to very low levels without strong effects on fuel consumption. These same data combined with powertrain analyses for an automobile indicate that early, preprototype systems could offer fuel economy comparable to 1975 prototype spark ignition automotive engines. Improvements currently being tested in the more advanced Rankine cycle automotive engines described in this report indicate that significant gains in fuel economy may be obtained. However, these gains have not yet been demonstrated in systems installed in vehicles. Further, the ultimate fuel economy obtainable after extensive development is not clear. The Rankine thermodynamic cycle, within practical limitations of pressure and temperature, is not one of inherently high efficiency compared to, for example, Otto or Diesel cycles. Current projections of fuel economy over the FDC for the early AAPS Program prototype systems are about 10 to 15 mpg. It is estimated that fully developed Rankine cycle automotive engines probably could provide very low pollution emissions with fuel economy approaching 12 to 20 mpg in full-sized cars over the FDC. The viability of the concept may well depend on the changing balance between emission and fuel consumption requirements and on the success of internal combustion engine designs in meeting these requirements.

From the standpoint of energy conservation and allocation in general, external combustion engines may be more attractive in the future because of the wider range of fuels that can be satisfactorily used in them. Efforts to conserve energy and to develop new energy sources, as shortages become more critical, may result in the availability of larger supplies of lower-grade fossil or synthetic fuels. External combustion engines may be the only automotive power plants that can operate efficiently with these fuels. Multifuel capability would probably be associated with an increase in emissions (particularly NO_x) over those reported herein for the AAPS Program, since the AAPS-developed combustors are predominantly of the vaporizing type. They consequently require a fuel volatility similar to that of gasoline.

The low power density of a Rankine cycle engine has not been completely eliminated in the present designs; thus, these engines are larger in size than other competing engine systems. Careful attention to packaging, however, should minimize the impact of engine size on vehicle aerodynamic drag and energy dissipation. It appears premature to assess the mass production cost of the Rankine cycle engine as the internal combustion engine, since the necessary manufacturing processes have not yet been defined.

In summary, then, the automotive Rankine cycle engine is still a leading candidate as a viable alternative to the spark ignition, internal combustion engine. Continued development efforts and supporting research are planned as long as there is evidence of satisfactory progress in improving system performance and approaching original program goals of low exhaust emissions without sacrificing competitive fuel economy levels.

5. STIRLING ENGINES

5.1 INTRODUCTION

5.1.1 General Description

The Stirling engine is an external combustion, closed cycle, piston-type power plant that uses a gaseous internal working fluid, usually hydrogen or helium (see Figures 5-1 and 5-2). Cyclical heating and cooling varies the pressure of the fluid within a closed volume; this pressure acts upon a piston, thereby developing output power. The heat transfer processes are accomplished at high efficiency by alternately and regeneratively moving the working fluid between heater and cooler sections. Thus, the engine contains at a minimum one expansion volume, one compression volume, one regenerator, one heater, and one cooler.

The ideal Stirling engine efficiency is 100 percent of the Carnot efficiency obtained between peak and minimum fluid temperatures. Various losses reduce the actual efficiency to about one-half of the ideal value. Nevertheless, the Stirling engine is one of the most efficient prime movers known.

5.1.2 Historical Background

The Stirling-cycle engine is not a recent development, having been patented in 1816 by Robert Stirling, a Scottish clergyman. Thousands of engines were built in the nineteenth century because they were substantially safer than existing steam engines, although slow and inefficient. They were eventually made obsolete by the development of the internal combustion engine and improvements in steam-engine boilers.

In 1938, N.V. Philips Laboratories at Eindhoven, Holland undertook the development of a modern version of the Stirling engine that could operate on a variety of fuels, originally as a means of generating electric power in remote areas. Although the market for such generators became very limited, continued research resulted in development of the rhombic drive displacer engine in 1953 and the roll-sock seal in 1960. This configuration had both the power piston and displacer piston in a single

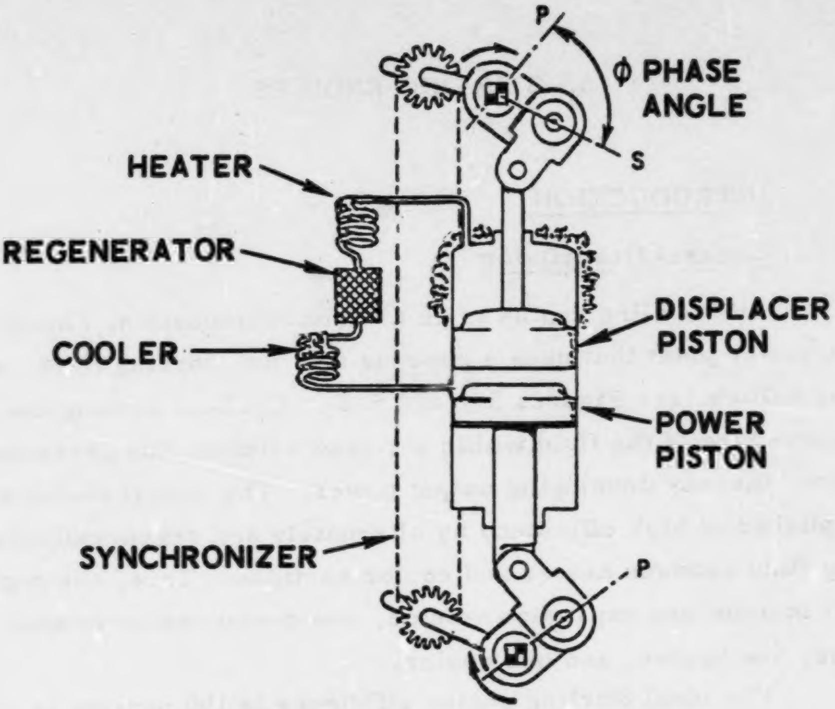

P
ϕ PHASE ANGLE
S
HEATER
REGENERATOR
DISPLACER PISTON
COOLER
POWER PISTON
SYNCHRONIZER
P

Figure 5-1. Basic Stirling Displacer Engine, Schematic

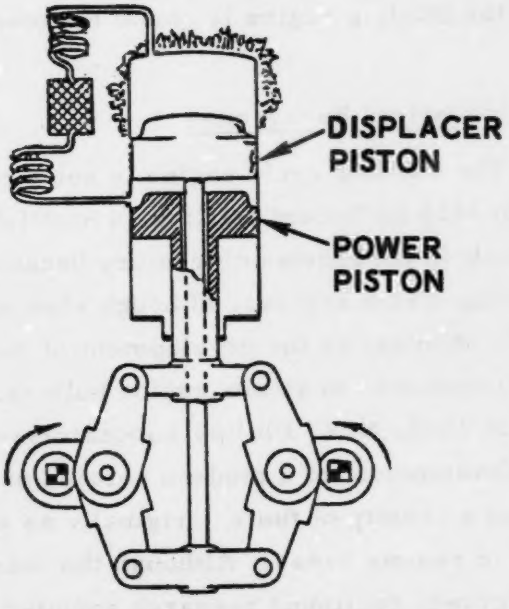

DISPLACER PISTON
POWER PISTON

Figure 5-2. Stirling Displacer Engine with Rhombic Drive, Schematic

cylinder. The rhombic drive made straightline rod movement possible while retaining the proper phase position between the pistons, allowed small-diameter seals to be used, and opened the way to high working pressure and high power. However, the most significant advancement in reducing engine volume and complexity occurred in 1970 at Philips with the successful operation of a double-acting Stirling engine with a swash-plate drive mechanism. The engine consists of four separate, interconnected cylinders in which the piston motion is phased at 90-degree intervals by the action of the swash plate. Each piston serves alternately as a power piston for one cylinder and as a displacer piston for the adjacent cylinder. This design represented a significant improvement in engine volume and complexity over other double-acting configurations.

5.2 POWER PLANT DESCRIPTION

The Stirling engine is a closed-cycle, external-combustion engine producing power by expanding a compressed volume of any type of heated gas. The Stirling cycle has a very high theoretical efficiency equaling Carnot cycle efficiency; actual efficiencies are about one-half of the theoretical values. The basic problems encountered in the development of this engine have been high weight, high volume, and packaging. But improvements have been made in recent years in reducing weight and volume by the use of double-acting pistons, which cut in half the number of pistons required (compared to the displacer engine), elimination the rhombic-drive gearing, and by the use of high pressure (200 atm) gaseous hydrogen as the working medium (Ref. 5-1). The thermal efficiency of the engine is improved in inverse relationship to the molecular weight of the working gas; low molecular weight has a further effect in reducing engine size.

A thermodynamic description of the Stirling cycle is given by Figure 5-3, which shows that the theoretical Stirling cycle is a closed regenerative cycle (heat stored and recovered) composed of two isothermal and two constant-volume processes with the regenerative action occurring during the constant-volume processes. The pressure-volume diagram shows a compression process from point (a) to point (b) where the piston does work

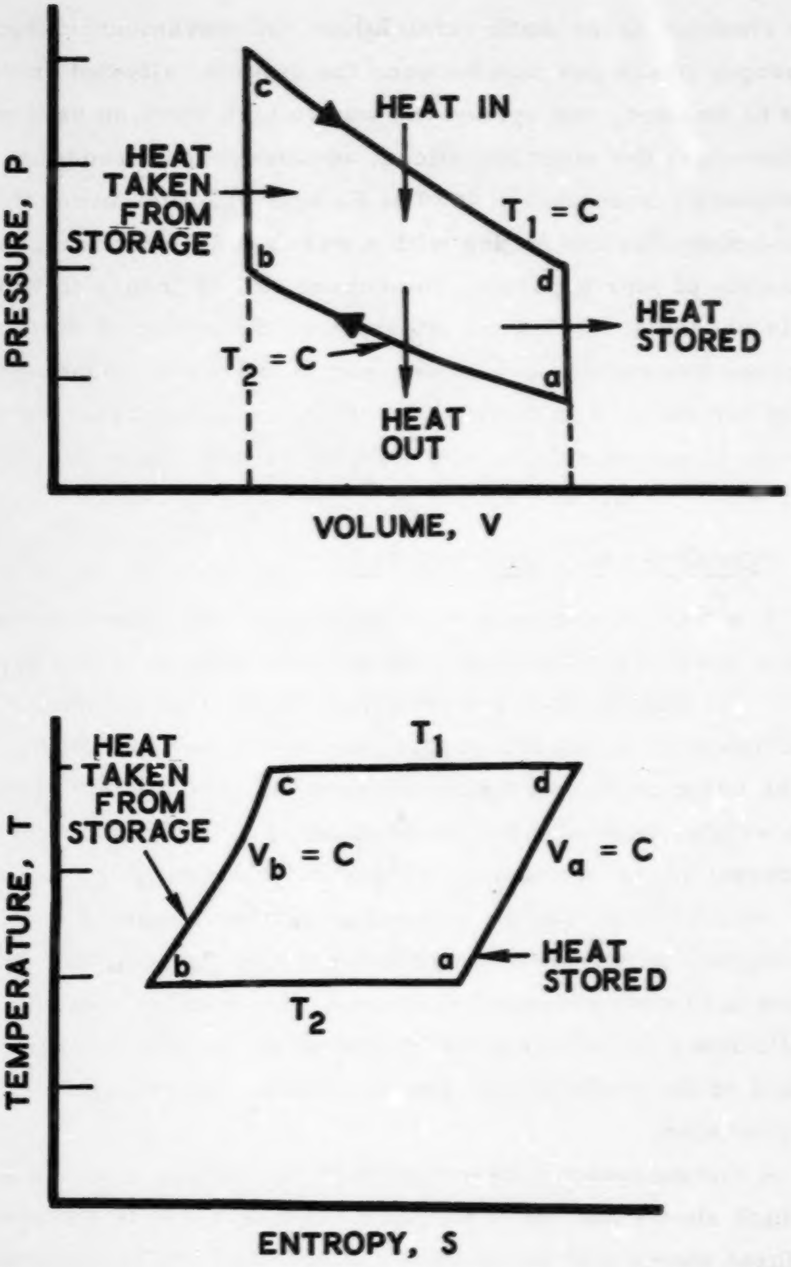

HEAT IN
c
HEAT TAKEN FROM STORAGE
$T_1 = C$
b
d
HEAT STORED
$T_2 = C$
a
HEAT OUT
PRESSURE, P
VOLUME, V
HEAT TAKEN FROM STORAGE
T_1
c
d
$V_b = C$
$V_a = C$
b
a
HEAT STORED
T_2
TEMPERATURE, T
ENTROPY, S

Figure 5-3. Thermodynamic Description of the Stirling Cycle

(Ref. 5-2)

compressing the gaseous medium at constant temperature as the cylinder volume decreases from its maximum value at point (a) to its minimum value at point (b). During the period (b) to (c) when the working-fluid volume is held constant, heat is transferred from an energy storage reservoir to the working medium thereby raising its temperature and pressure to maximum values at point (c). At this point the expansion stroke begins and continues to point (d). During this stroke the working medium does work on the piston at constant temperature as the cylinder volume increases from its minimum value at point (c) to its maximum value at point (d). At the end of the expansion stroke the working fluid is held at constant volume while heat is transferred to the storage reservoir. During this process the working fluid temperature and pressure drop to minimum values.

If regeneration is 100 percent efficient, the amount of heat stored during the constant volume process following expansion is equal to the heat taken from storage during the constant volume process after compression. Also, an external combustion process is utilized to transfer heat during the expansion process to maintain a constant temperature T_1. During the compression process a radiator or heat exchanger transfers heat out into the environment to maintain the working medium at a constant temperature T_2.

5.2.1 Power Plant Configuration

The basic components of the early Stirling displacer engine are shown in Figure 5-1 and consist of a displacer piston, power piston, heater, cooler, and regenerator. The motion of the displacer piston is synchronized with the suction of the power piston so as to produce the required pressure/volume relationships for the Stirling cycle in conjunction with periodic cooling and heating of the gas by the heater, cooler, and regenerator.

The rhombic drive, illustration in Figure 5-2, was a development accomplished in 1953 which provided a means for actuating the displacer piston in proper phase relation with the power piston, both operating in single cylinder. An additional feature of the rhombic drive unit is that it contains a buffer space which allows the crankcase to remain at atmospheric pressure. Mass motions in the design result in perfect dynamic balance of a one-cylinder engine.

Figure 5-4 shows the efficiency of a 225-hp per cylinder rhombic drive engine with air, helium and hydrogen as alternative gaseous working mediums. As shown, high power levels combined with high efficiencies (of the order of 40 percent) are achievable (particularly true for the hydrogen working medium). This was the most common form of Stirling engine design under development between the years 1953 and 1971 by the Philips Company and its licensees, General Motors Corporation, United Stirling of Sweden, and M.A.N./MWM of Germany. As shown in schematic form in Figure 5-5, the rhombic-drive, displacer-type engine has single acting pistons arranged in pairs and linked together through a common drive mechanism. One is called a displacer piston and the other is called a power piston. The rhombic drive consists of two meshed timing gears, two crankshafts, two power-piston connecting rods, and a displacer-piston yoke. Heat is added by an external combustor to the heater tubes connected to the cylinder head on one side of the displacer piston. The water-cooled cooler tubes are connected to the other side of the cylinder between the displacer piston and the power piston. The heater tubes and cooler tubes are connected to each other through the regenerator, which usually consists of a small volume filled with a fine-wire matrix. At any instant the working-fluid pressure is essentially uniform, since the working fluid spaces (formed by the two pistons) are connected to each other through the cooler tubes, regenerator, and heater tubes.

In operation, the motion of the power piston lags the motion of the displacer piston by about one quarter of a revolution. When the power piston is in the downward position and moving very slowly, the displacer piston is moving upward and rather rapidly. This forces the bulk of the working fluid around through the heater tubes, regenerator, and cooler tubes to the cold space between the displacer and power pistons. In so doing, the bulk of the fluid is cooled as heat is transferred to the regenerator and the fluid pressure drops. The power piston then moves upward, compressing the fluid, increasing its pressure, and transferring heat from the bulk of fluid to the cooler tubes. The displacer then moves downward, moving

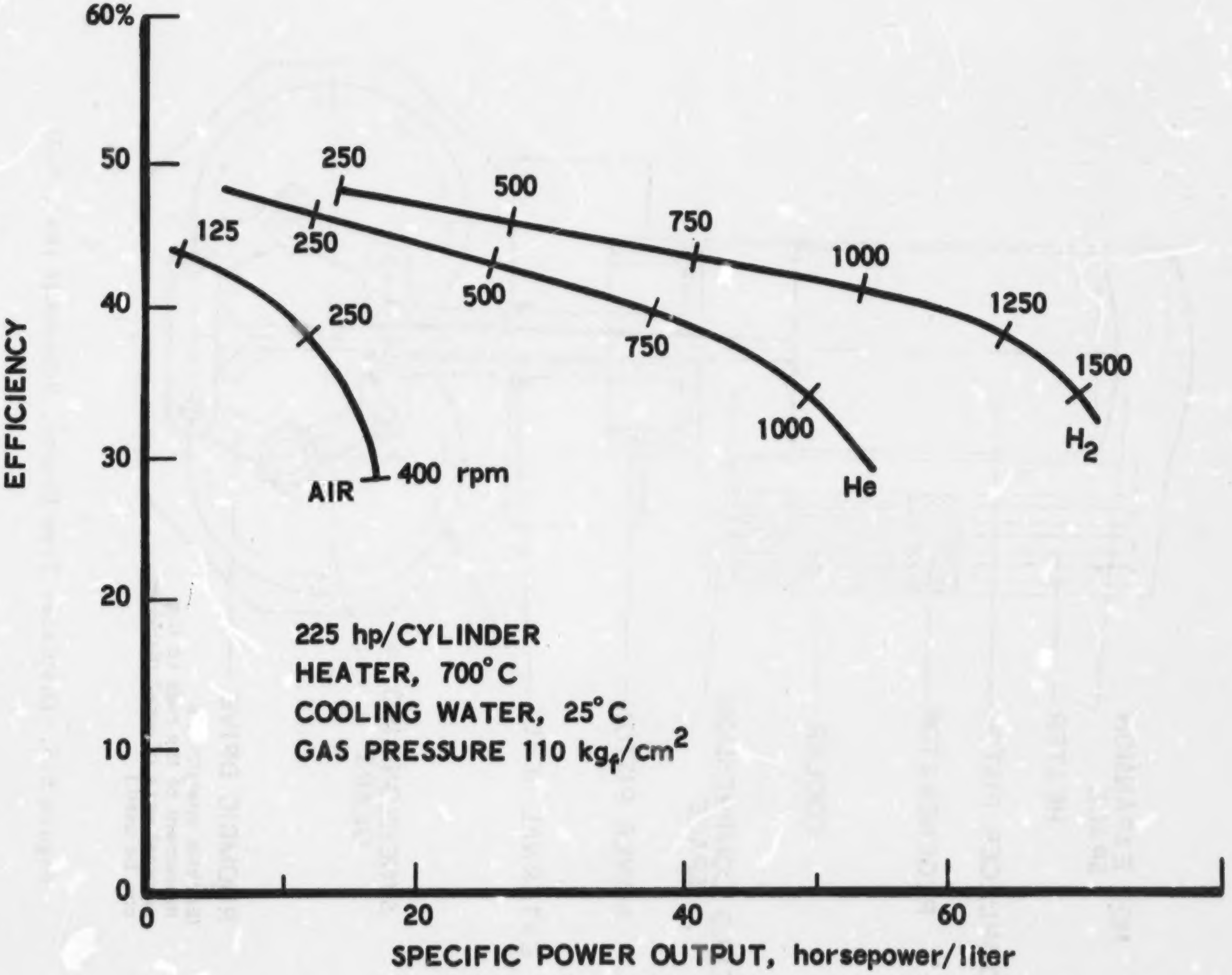


Figure 5-4. Stirling Engine Efficiency versus Specific Power Output (Ref. 5-3)

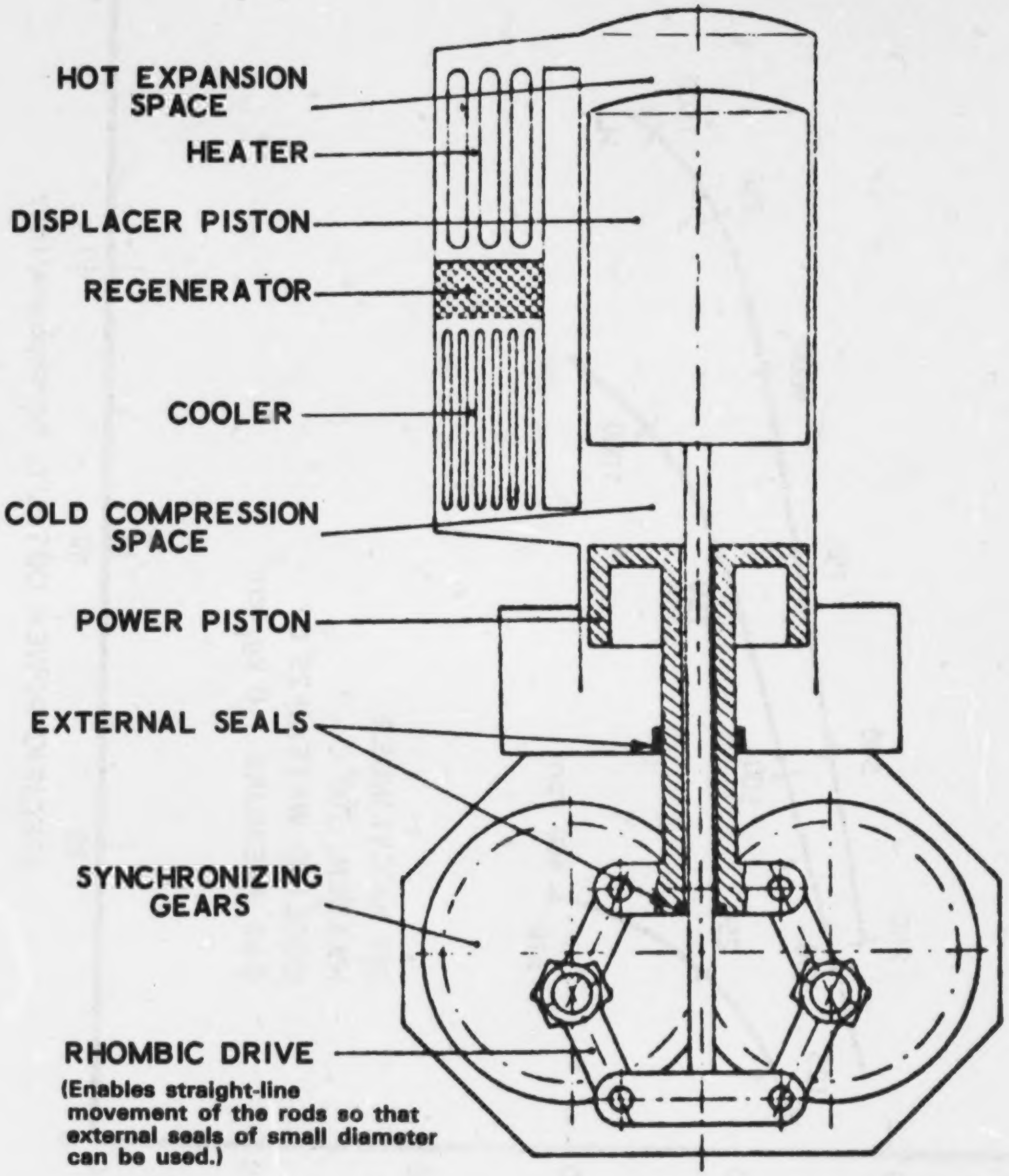

HOT EXPANSION SPACE
HEATER
DISPLACER PISTON
REGENERATOR
COOLER
COLD COMPRESSION SPACE
POWER PISTON
EXTERNAL SEALS
SYNCHRONIZING GEARS
RHOMBIC DRIVE
(Enables straight-line movement of the rods so that external seals of small diameter can be used.)

Figure 5-5. Displacer Type Engine, Schematic (Ref. 5-3)

the bulk of the fluid back through the regenerator where heat is transferred, thereby further increasing its pressure. The power piston then moves downward as the work fluid expands and moves the bulk of fluid through the heater tubes, thereby effecting a heat transfer and completing the cycle. These motions do not occur separately but are brought about by the approximately 90-degree, out-of-phase motion of the two pistons.

Because the internal fluid pressure is low when the power piston moves upward and high when the piston moves downward, a net work output is produced. The effect of the regenerator is to improve efficiency by storing heat from the hot fluid as it flows toward the cold space and by releasing the heat when the flow reverses.

Fuel is burned in the combustion chamber in the vicinity of the heat transfer tubes, and the combustion gases pass through this array of tubes surrounding the combustion chamber. The exhaust gases leave at high temperature and pass through the preheater, which cools the exhaust and warms the incoming combustion air. This technique conserves heat that would otherwise be wasted.

The most significant advancement in reducing engine cost and volume has come about through application of the double-acting Stirling engine concept and swash-plate drive mechanism. An engine of this design was run at the Philips Laboratories in 1970 (Ref. 5-3). Shortly thereafter, the Ford Motor Car Company entered into an agreement with Philips to develop a swash-plate drive, four-cylinder, double-acting Stirling engine for automobile application. (See Figures 5-6 through 5-9.) A picture of the engine mockup for a Ford Torino automobile is shown in Figure 5-9. Preheater and burner are at the front. Exhaust pipes show at bottom, with an exhaust gas recirculation tube extending upward to an intake-air blower. A combined water pump and power steering pump, belt-driven, is located near the front, and the alternator is shown at the rear corner.

United Stirling of Sweden is also developing a four-cylinder, double-acting Stirling Engine with a conventional crankshaft for another

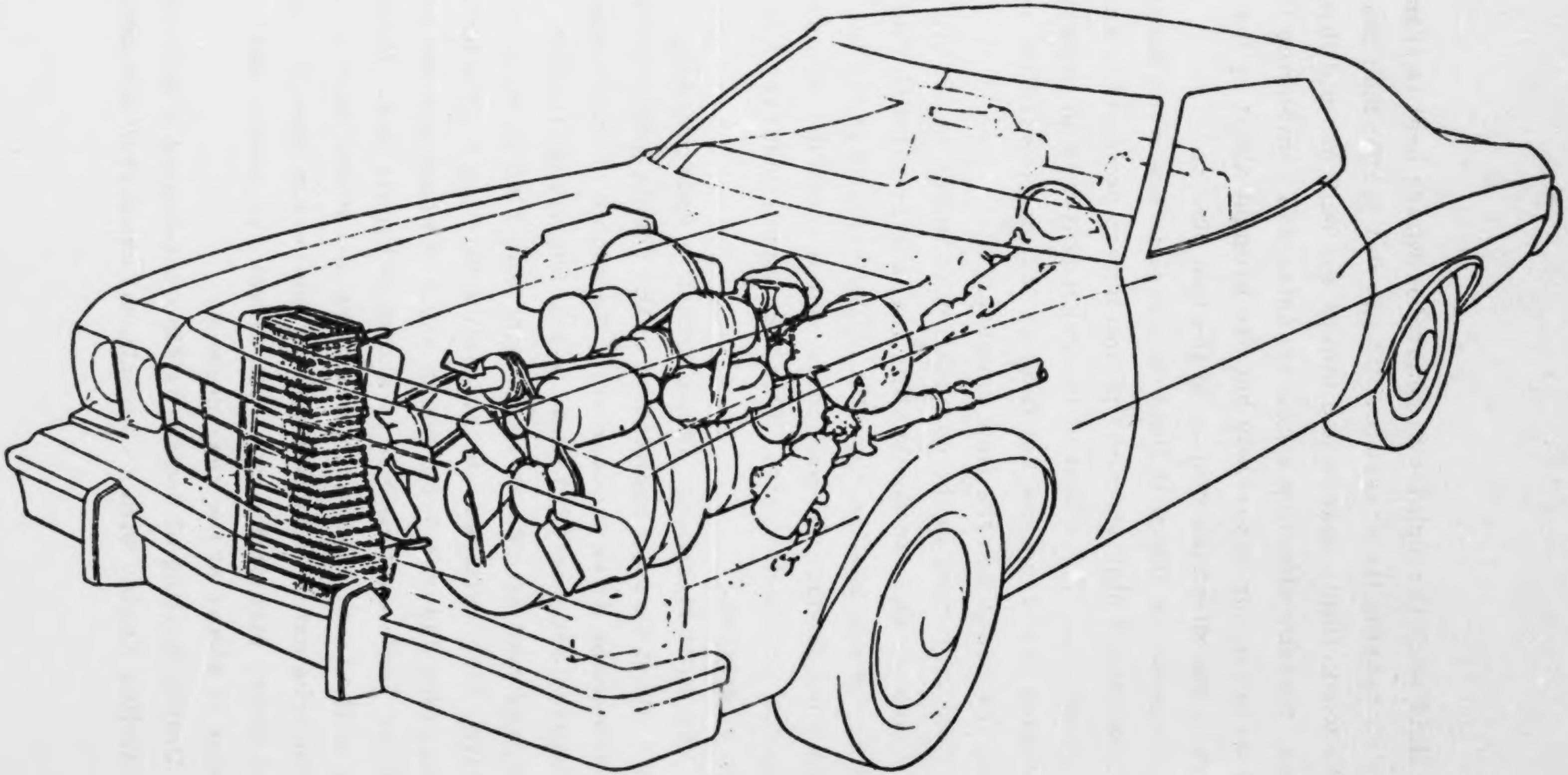


Figure 5-6. 1973 Ford Torino with Philips Stirling Double-Acting Engine (Ref. 5-13)

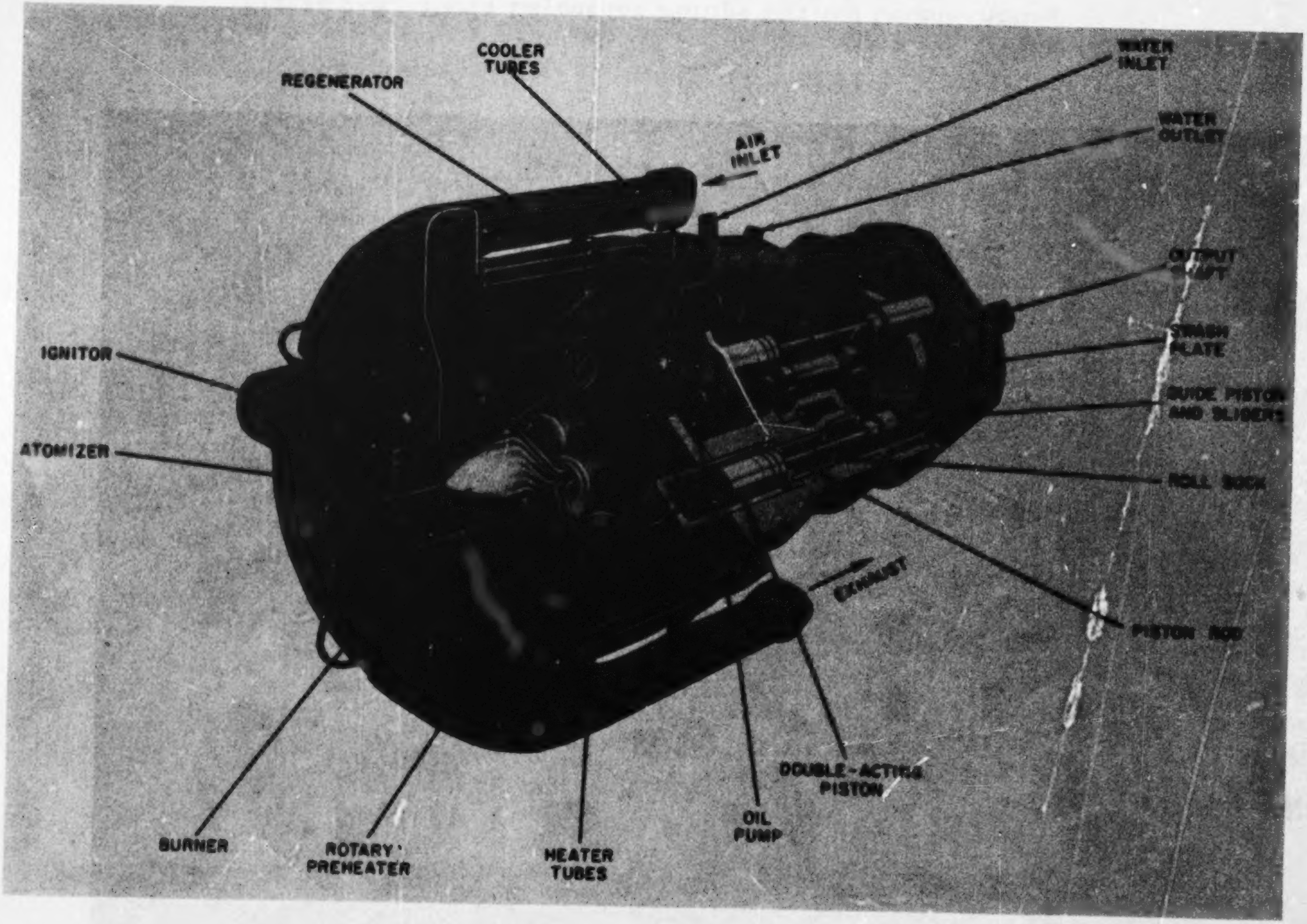

COOLER TUBES
REGENERATOR
AIR INLET
WATER INLET
WATER OUTLET
OUTPUT SHAFT
SWASH PLATE
GUIDE PISTON AND SLIDERS
ROLL SOCK
PISTON ROD
EXHAUST
IGNITOR
ATOMIZER
BURNER
ROTARY PREHEATER
HEATER TUBES
OIL PUMP
DOUBLE-ACTING PISTON

Figure 5-7. Philips Stirling Double-Acting Engine with Swash-Plate Drive (Ref. 5-13)

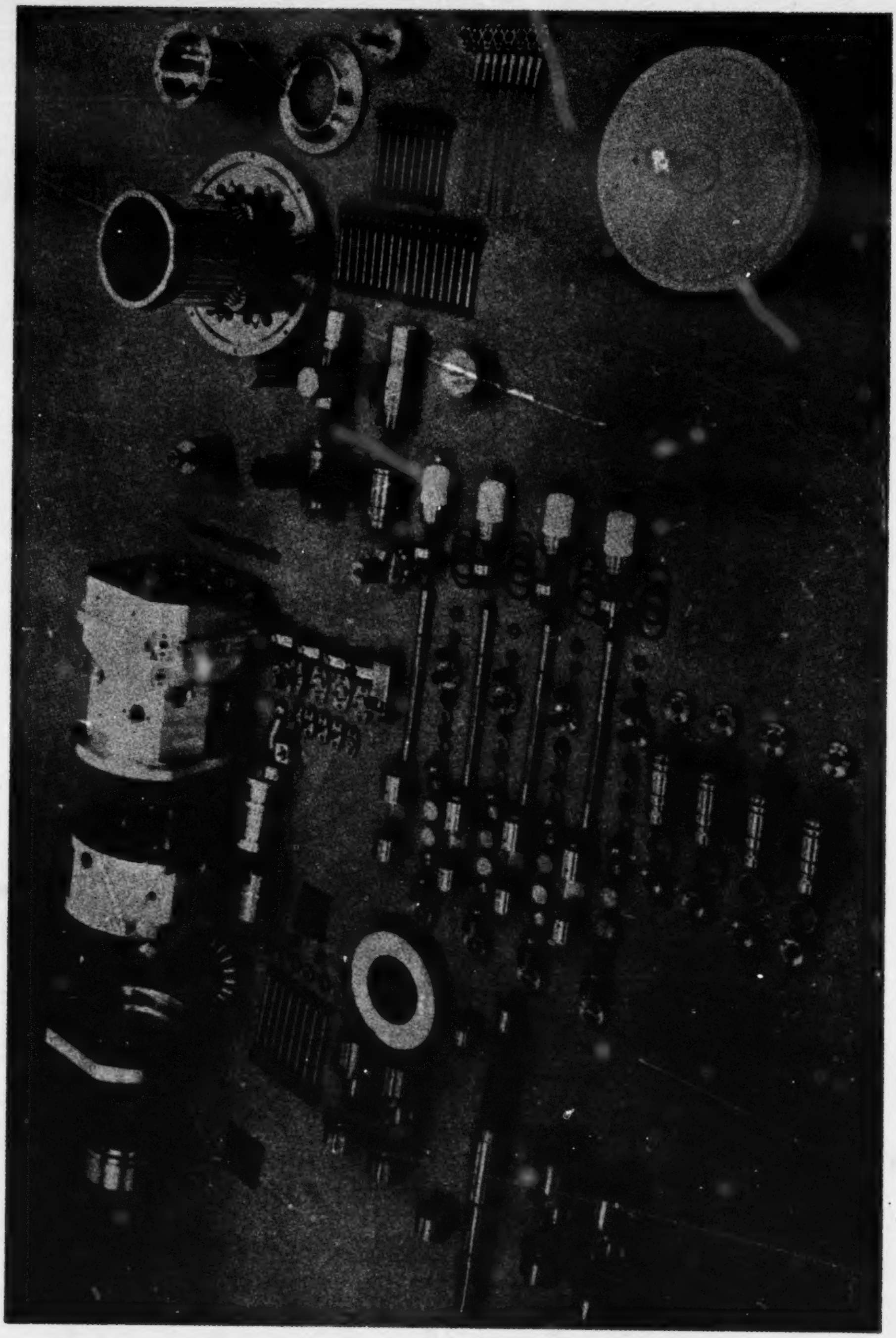


Figure 5-8. Parts Layout for Philips Stirling Double-Acting Engine, Torino Application (Ref. 5-13)

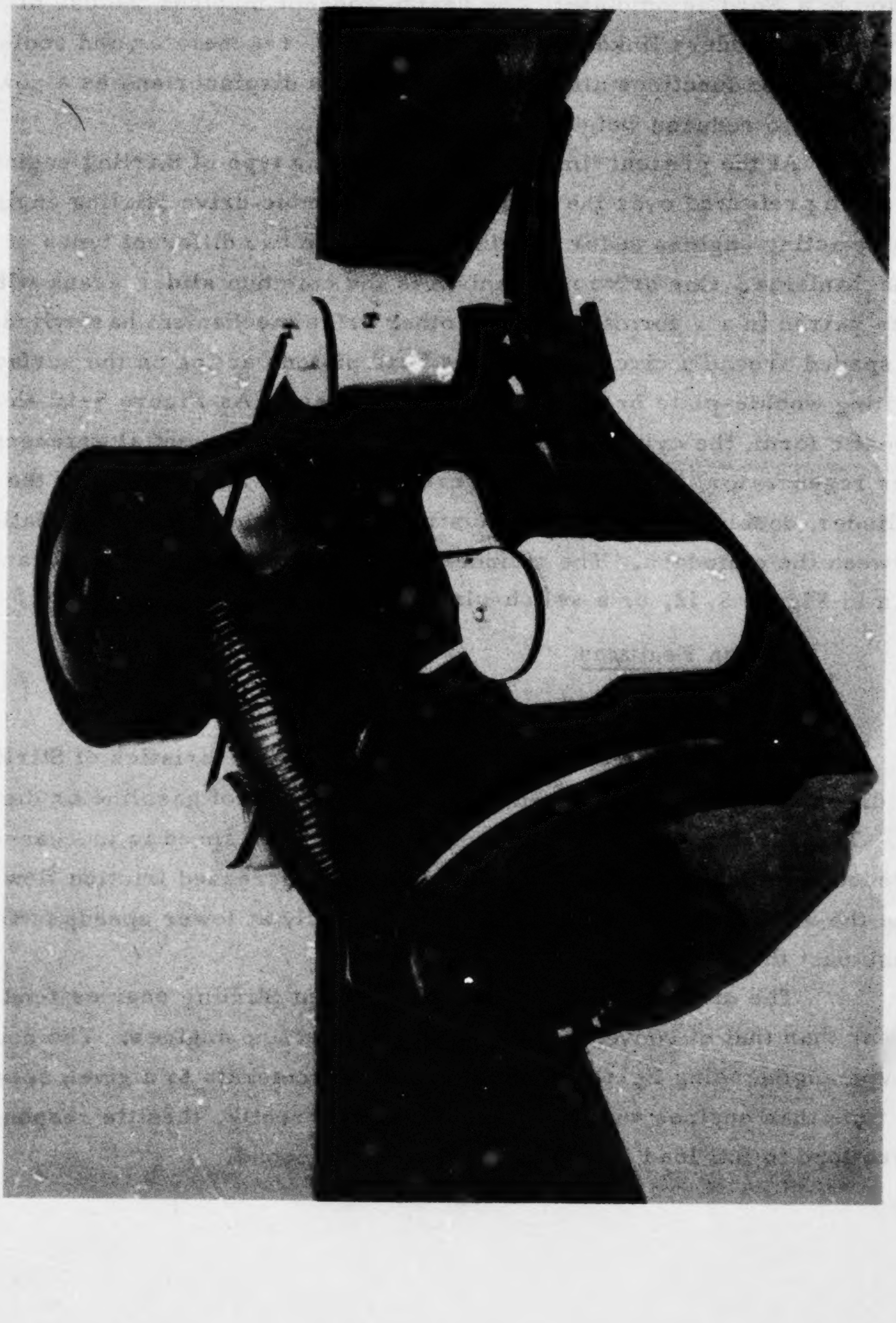


Figure 5-9. Front 3/4 View of Philips Stirling Double-Acting Engine Mock-Up for a Ford Torino (Ref. 5-13)

application to a Ford automobile. The design concept involves the use of four pistons in cylinders linked in series by heater, regenerator, and cooler units. Each piston functions alternately as a piston displacer and as a power piston, leading to reduced weight and volume.

At the present time, the double-acting type of Stirling engine is very much preferred over the single-acting, rhombic-drive Stirling engine. The double-acting engines under development utilize two different types of drive mechanisms. One drive mechanism is the common slider crank with cylinders paired in a V formation. The other drive mechanism has cylinders equally spaced around a circle with cross-head pistons acting on the surfaces of a rotating wobble-plate or swash-plate mechanism. As Figure 5-10 shows in schematic form, the cylinders are interconnected by sequential arrangement of heater regenerator, and cooler. Figure 5-11 shows a schematic of the four-cylinder, double-acting engine, illustrating the 90-degree phase relationship between the cylinders. The cylinders could be connected to a V-crank, as shown in Figure 5-12, or a swash-plate mechanism shown in Figure 5-13.

5.2.2 Design Features

5.2.2.1 Power

As shown in Figure 5-14 the torque characteristics of Stirling engines droop as speed is increased, somewhat like that of gasoline or diesel engines. The Stirling engine torque droops naturally as speed is increased due to reduced effective heat-transfer capacity and increased friction flow losses of the working fluid. Torque drops off rapidly at lower speeds owing to leakage past the power piston.

The maximum output speed of current Stirling engines tends to be lower than that of conventional automotive gasoline engines. The double-acting type engine being lighter in weight should accelerate to a given speed in shorter time than engines with rhombic drive. Currently, throttle response from zero load to full load is approximately 0.30 second.

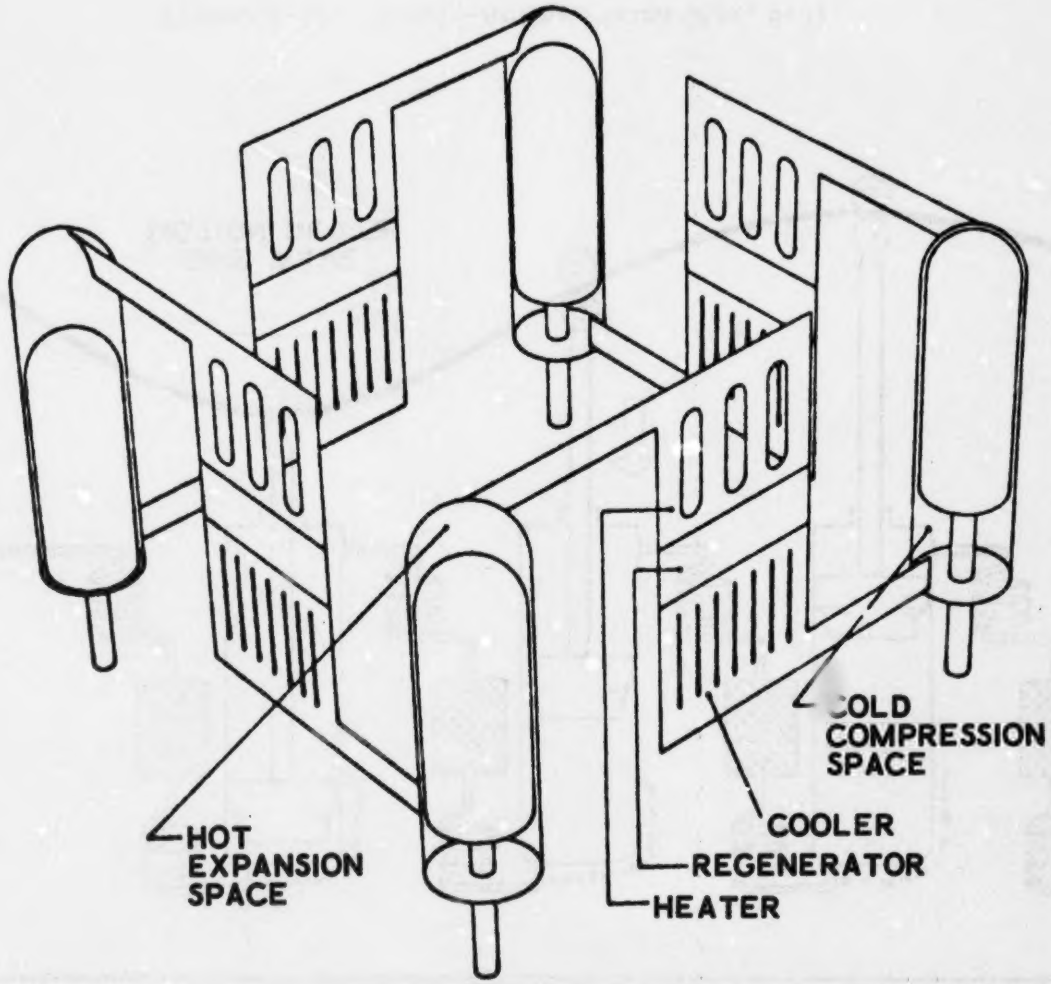

COLD
COMPRESSION
SPACE
COOLER
REGENERATOR
HEATER
HOT
EXPANSION
SPACE

Figure 5-10. Double-Acting Type Engine, Schematic (Ref. 5-3)

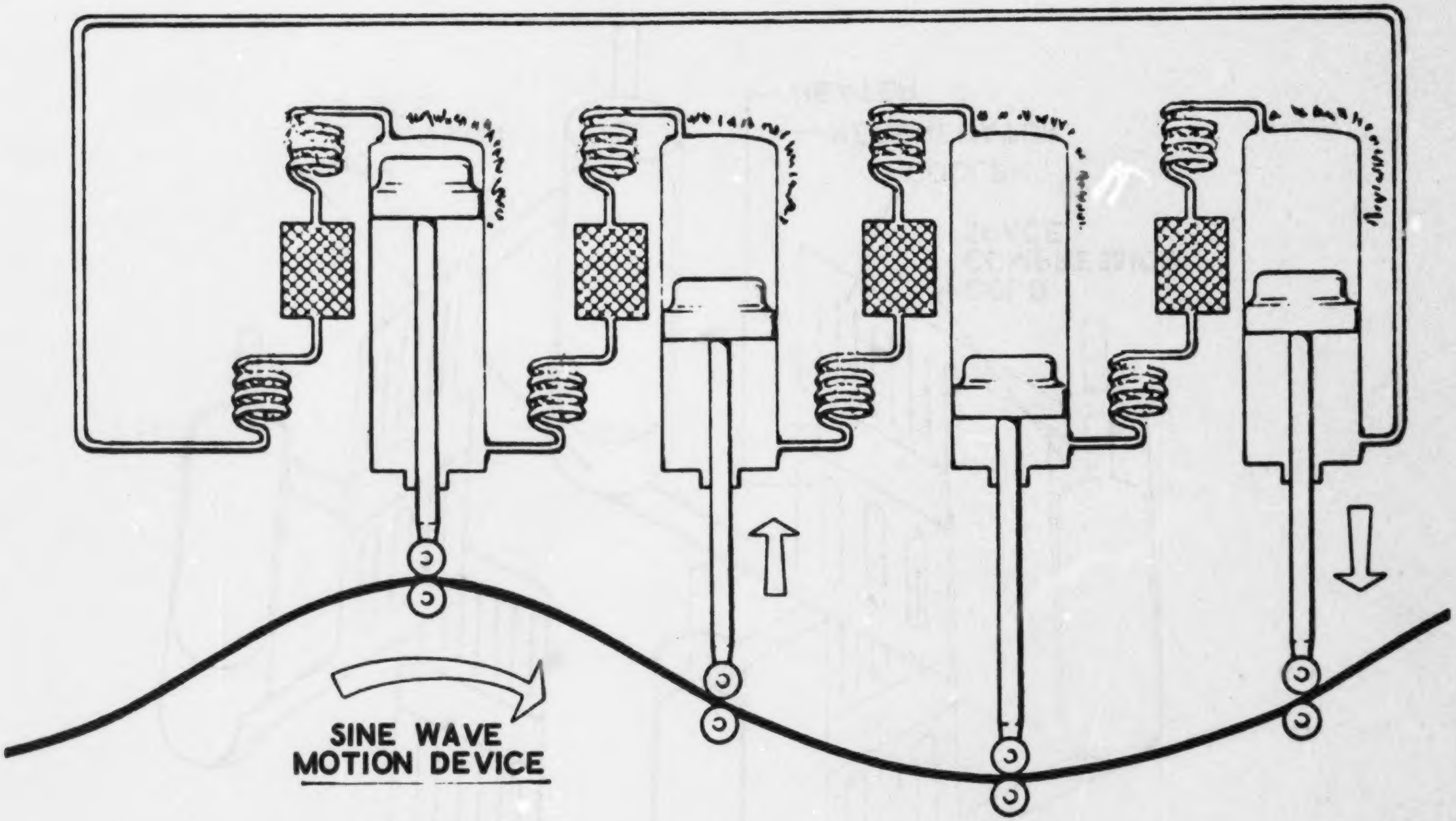

SINE WAVE
MOTION DEVICE

Figure 5-11. Double-Acting Piston (Ref. 5-1)

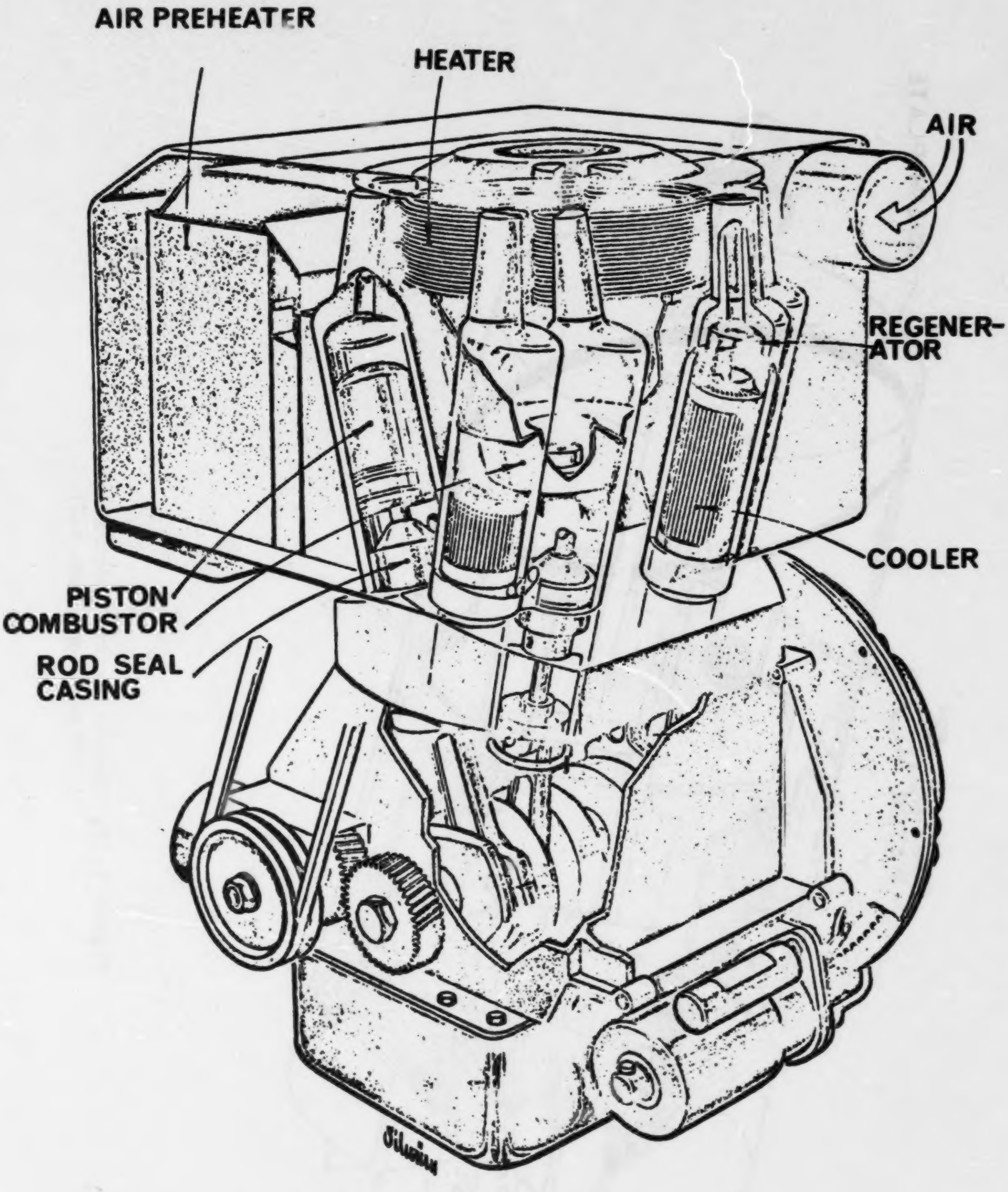

AIR PREHEATER
HEATER
AIR
REGENERATOR
COOLER
PISTON
COMBUSTOR
ROD SEAL
CASING
Silvain

Figure 5-12. Double-Acting V-4 Engine Preprototype (Ref. 5-4)

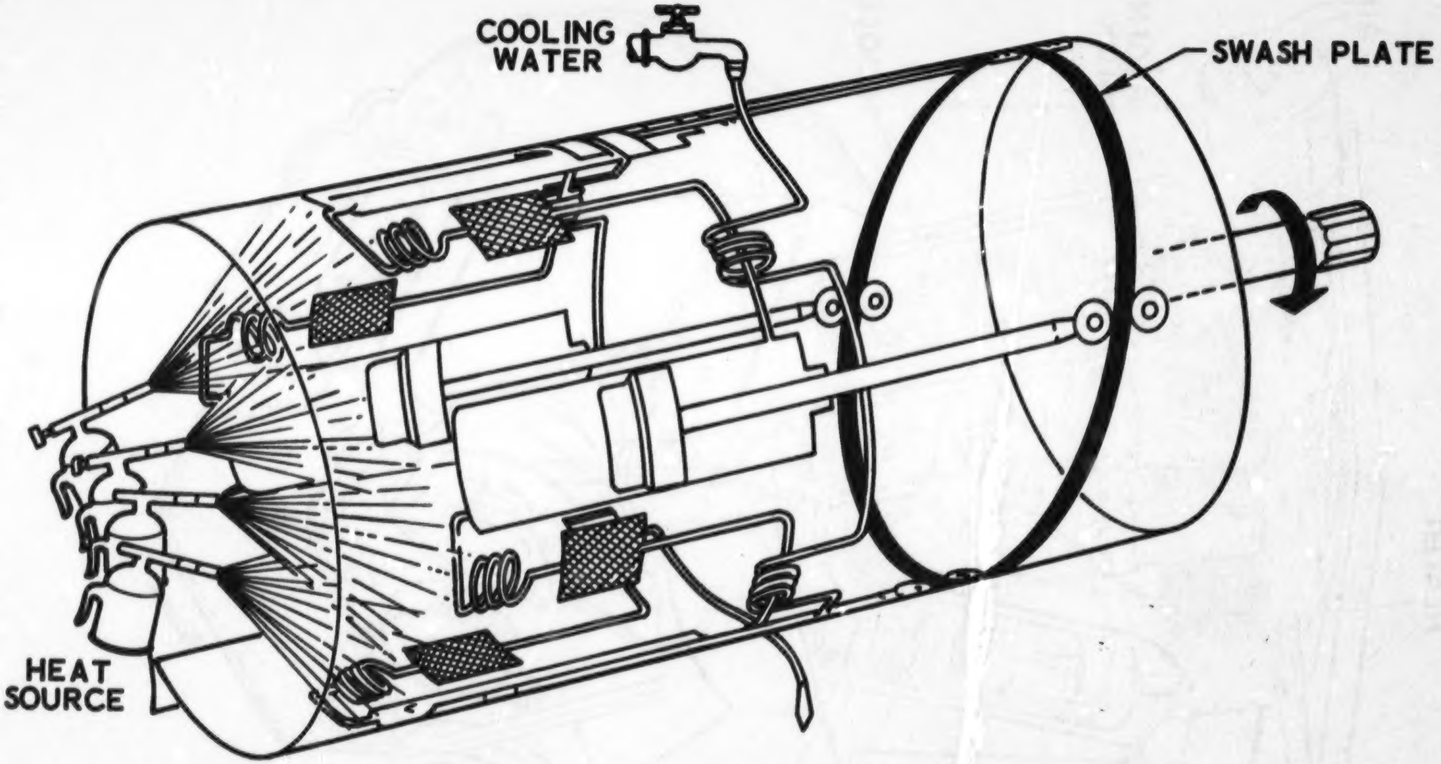

COOLING WATER
SWASH PLATE
HEAT SOURCE

Figure 5-13. Double-Acting Cylindrical Configuration (Ref. 5-5)

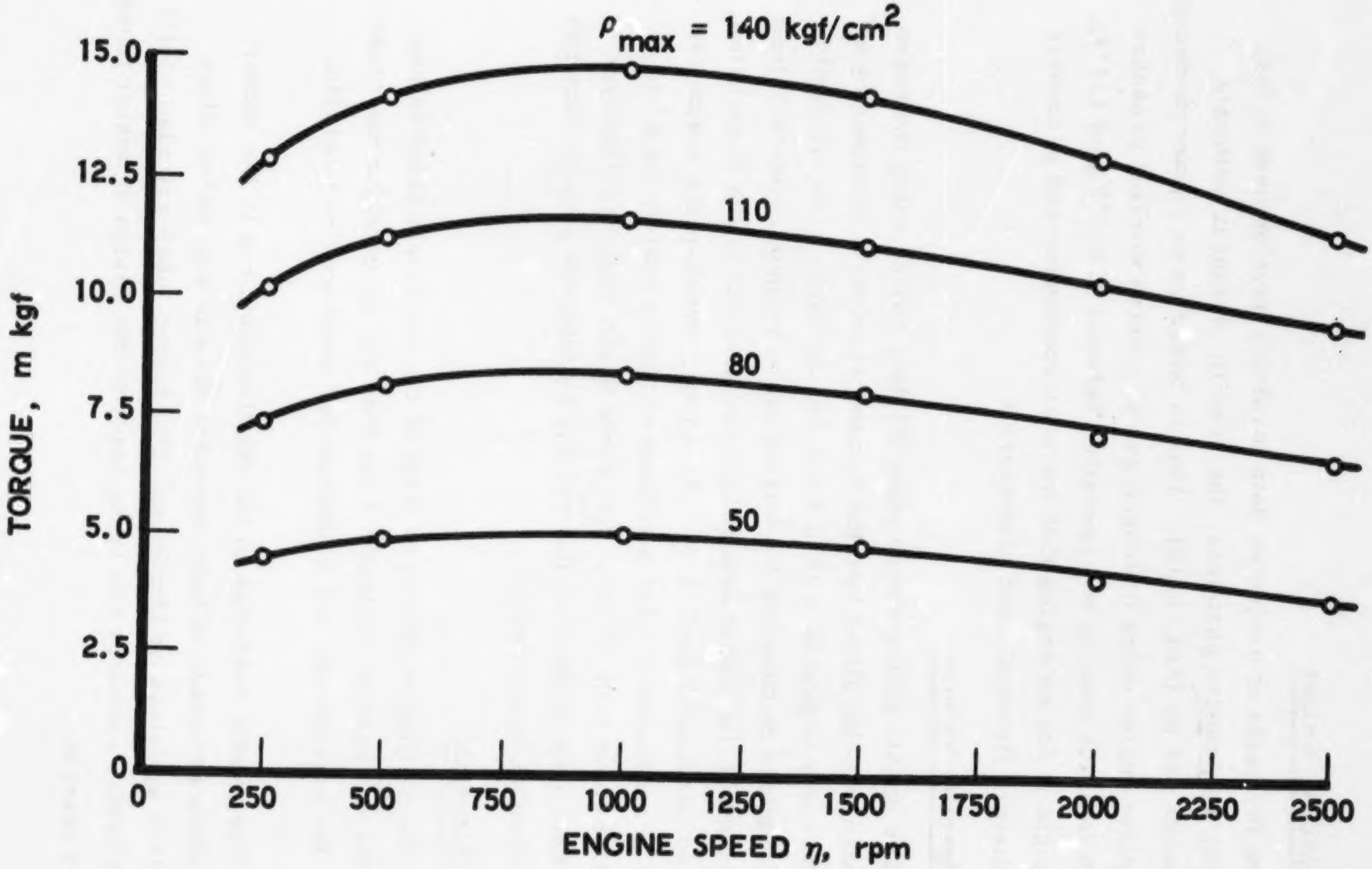


Figure 5-14. Dynamometer Torque of a 40-hp Stirling Engine as a Function of the Speed η at Different Values of P max (Ref. 5-2)

5.2.2.2 Specific Weight

On the basis of projected data involving development of the Stirling engine for automotive purposes, the specific weight is estimated as 3.5 to 4.5 pounds per hp (Ref. 5-12). This is based on an advanced-design 170-hp, swash-plate engine using hydrogen gas at a mean working pressure of 3,000 psi with indirect heating and operating between 1,380°F and 131°F. This specific weight is for an engine that includes combustor and preheater but excludes radiator, flywheel, and accessories.

5.2.2.3 Specific Volume

The double acting swash plate Stirling engine using hydrogen gas shows promise of being fitted into the volume previously occupied by a conventional Otto cycle engine in a 1972 Ford Torino (Ref. 5-5). A number of changes are required in coupling the engine to the transmission (a 5-inch rearward movement of the transmission for example) and in the suspension, steering box, and dashboard (Ref. 5-5). A 170-hp, swash-plate engine for this application is estimated to have a volume of approximately 10 ft^3 or a specific volume of about 0.06 ft^3/hp. Recourse to the double-acting type of Stirling engine has reduced the specific volume by a factor greater than two compared to a rhombic-drive engine.

5.2.2.4 Materials

The Stirling engine uses most of the materials found in conventional internal combustion engines. Cast iron can be used for the crankcase, steel for the crankshaft, and aluminum for most non load carrying members.

The costly material in the Stirling engine is in the heater tubes. These tubes are made of heat resistant chrome and nickel alloys such as SAE 30310 stainless or Multimet. The heater tubes are also costly to fabricate. Further advances are being made in the design of heater tubes to improve heat transfer.

Several developers are testing the heat-pipe concept for use in this engine. The concept makes use of an enclosed volume with a porous lining filled with a heat-carrying-fluid medium such as sodium. At the hot end of the heat pipe sodium is vaporized and allowed to flow to the cool end, where it condenses (Ref. 5-6); it returns to the hot end by capillary action. The aforementioned improvements permit the heat tubes to be shorter, thereby lowering friction losses, increasing efficiency, and reducing the cost of materials in the heater head. Also, the distribution of temperature is improved with the use of sodium thereby lengthening the life of the heater head.

It must also be noted that the use of the double acting principle permits one heater head to be used for a multicylinder engine. This cuts down the cost of fabricating heater heads, reduces heater head size, and makes the engine more compact for automotive purposes.

5.2.2.5 Fuel Requirements

The Stirling engine is an external combuston engine and, therefore, a multifuel engine. It can operate with any of the commercial fuels sold on the open market and has no special octane or cetane requirements. Because the fuel is injected into the combustor, it is necessary that the fuel be filtered to prevent clogging of the injector orifice. Commercial fuels are highly filtered and, therefore, are preferred to noncommercial fuels.

5.2.2.6 Temperatures and Pressures

The Stirling engine must be operated at high heater temperatures and relatively low cooler temperatures if high efficiency is to be achieved. Whereas the maximum cycle temperature of an Otto cycle engine will reach 3,500 to 4,000°F, the Stirling engine operates at a lower temperature. For example, the Ford Torino engine is designed for 1,470°F operation (Ref. 5-5). This is the practical operating temperature for heater tubes fabricated from heat-resistant and creep-resistant stainless steels.

5.2.2.7 Working Fluids

The preferred working fluid is hydrogen gas. This gas has properties conducive to low friction losses and high heat transfer. Under high working pressures and temperature, hydrogen can diffuse through the metal container wall and thereby escape out into the atmosphere. A thin coating of silicon nitride is used as a hydrogen gas diffusion barrier. It has been claimed that the coating is effective under apparently static internal pressurization conditions. It is not known whether the coating effectiveness varies for materials undergoing deformation during creep and fatigue conditions.

Helium gas is the next preferred fluid. It has low molecular weight and does not diffuse through container walls. Helium is safe to handle under both low and high pressure. However, thermal efficiency of the engine is lower than that of the same engine using hydrogen gas as the working medium (Figure 5-4). Air was used at one time, but the thermal efficiency was lower than for either hydrogen or helium.

5.2.2.8 Cooling Requirements

Compared to an Otto or Diesel Cycle engine, the Stirling engine transfers a much greater proportion of its heat through its coolant. For this reason, the Stirling engine in certain designs may require a radiator about 2.5 times as large as that for an internal combustion engine of comparable power. The following table shows percentages of heat balances for Stirling and diesel engines (Ref. 5-8).

	Diesel, %	Stirling, %
Exhaust	35	9
Cooling Water	19	47
Friction	7	4
Auxiliaries	2	4
Effective	37	36
	100	100

EPA's research and development program in the Rankine cycle power plant field has resulted in design and fabrication of a steam condenser of smaller size than previously available for air cooled systems (Ref. 5-9). This means that the radiator for a Stirling engine can be reduced to a size compatible with automobile engine compartments by using using the ideas involved in design of the aforementioned steam condenser. It should be noted that the Philips Company has also developed, and is further improving, a smaller size radiator for Stirling engine automobile application (Ref. 5-3). This radiator is called a "Folded Frontal Area Radiator" and consists of fine-gauge metal attached to small-diameter, coolant-carrying tubes folded like an accordion to present a large frontal area within a small volume.

The cost to manufacture the two types of aforementioned radiators is likely to be higher than that for conventional-type radiators found in present-day automobiles.

5.2.2.9 Transmission

The engine can use a conventional transmission, either manual or automatic. A properly designed engine could provide high torque at low speed with a relatively flat characteristic over the speed range. On this basis, it might be anticipated that fewer gear changes would be required, compared to an Otto cycle engine.

5.2.2.10 Lubrication

Because the Stirling engine is an external-combustion, low-temperature engine, the lubrication oil does not operate in an environment as hot as that found in the Otto cycle engine. Use of helium or hydrogen gas requires special seals and/or use of a crosshead piston designed to restrict the flow of oil into the hot-working-fluid area or flow of the working fluid into the crankcase. The oil consumption has been found to be minimal in a Stirling engine, and it would be advantageous to use the same oil in the transmission as in the engine.

5.2.2.11 Filtration

Filtration of combustion air is required to prevent fouling and clogging of air passageways formed by the stacking of heater tubes. The filter must be able to handle air at low pressure and at high preheat temper-temperature.

Filtration of lubricating oil is required to prevent access of abrasive materials to lubricated load carrying surfaces such as the bearings and cylinder surfaces.

5.2.3 Operating Characteristics

5.2.3.1 Power Control Requirements and Methods

The output of a Stirling engine of given displacement is nearly a linear function of the mean pressure of the working gas. Sudden changes in mean pressure can effect sudden changes in engine output. This fact forms the principle of control.

The Philips control system consists of two independent operations: Fuel flow is controlled by the heat exchanger exit temperature, which is affected by the pressure of the working fluid and heat input from the combustor. Engine throttling is controlled by varying the pressure in the chamber housing the working fluid.

A schematic of the pressure-regulating method is shown in Figure 5-15 where a reservoir supplies high pressure gas to the working fluid space. Throttle control is effected by opening and closing the supply and short-circuit valves as required. With this system a compressor is required to pressurize the gas and return it to the reservoir.

The power control system schematic is shown in Figure 5-16 and the fuel control system in Figure 5-17. With this method of control it is possible to go from zero power to full power in 0.3 seconds.

5.2.3.2 Starting Characteristics

The Stirling engine has reliable starting qualities under all conditions. Startup time from the cold condition is determined by design

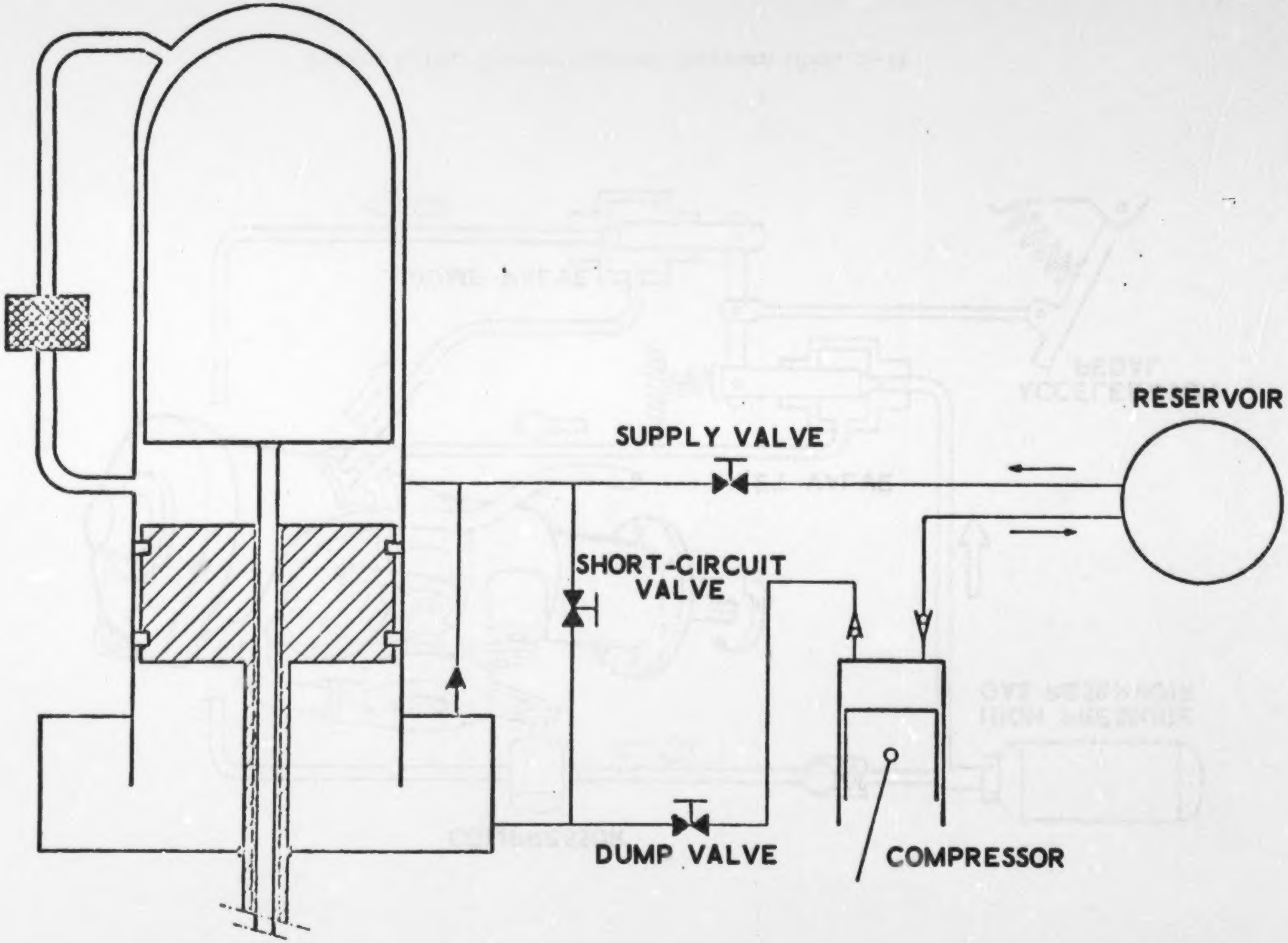

RESERVOIR
SUPPLY VALVE
SHORT-CIRCUIT VALVE
DUMP VALVE
COMPRESSOR

Figure 5-15. Schematic of Pressure-Throttling Method (Ref. 5-3)

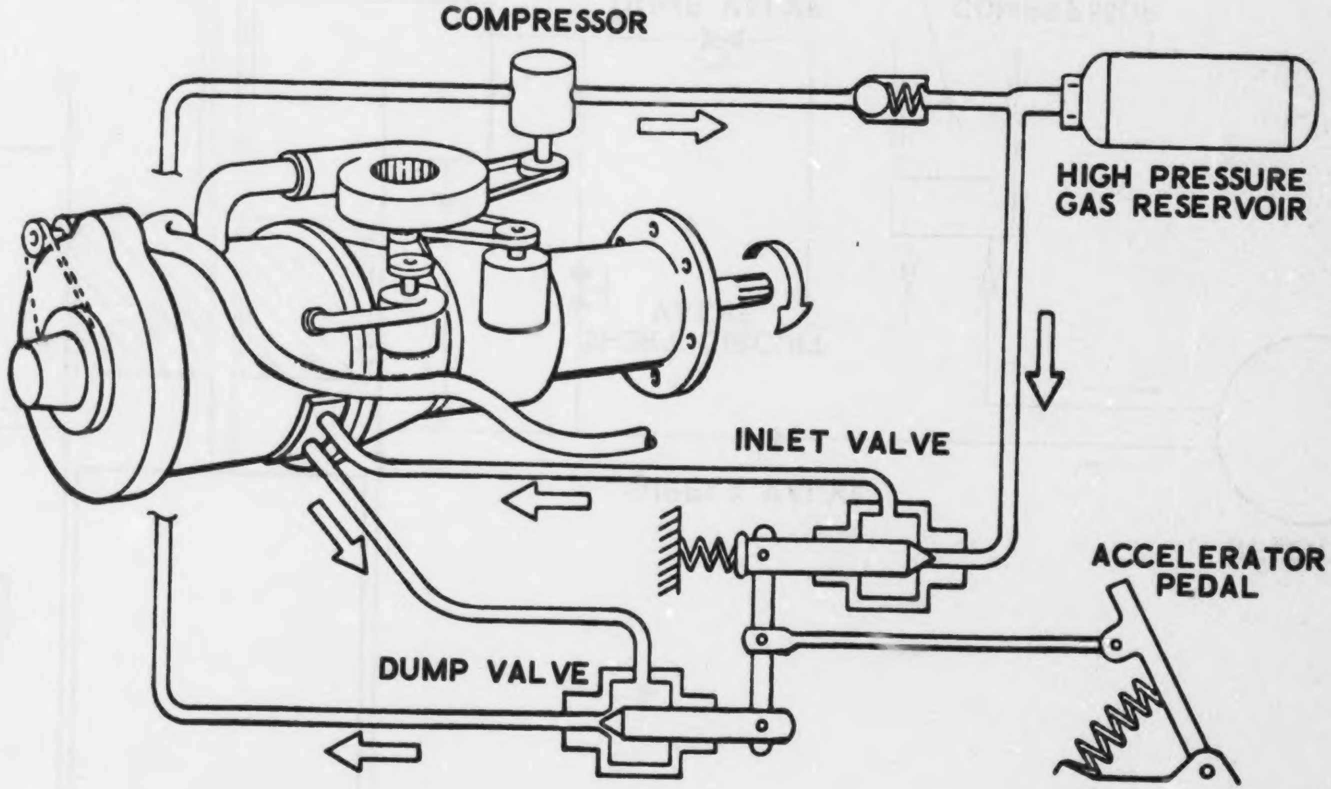

COMPRESSOR
HIGH PRESSURE GAS RESERVOIR
INLET VALVE
ACCELERATOR PEDAL
DUMP VALVE

Figure 5-16. Power Control Diagram (Ref. 5-1)

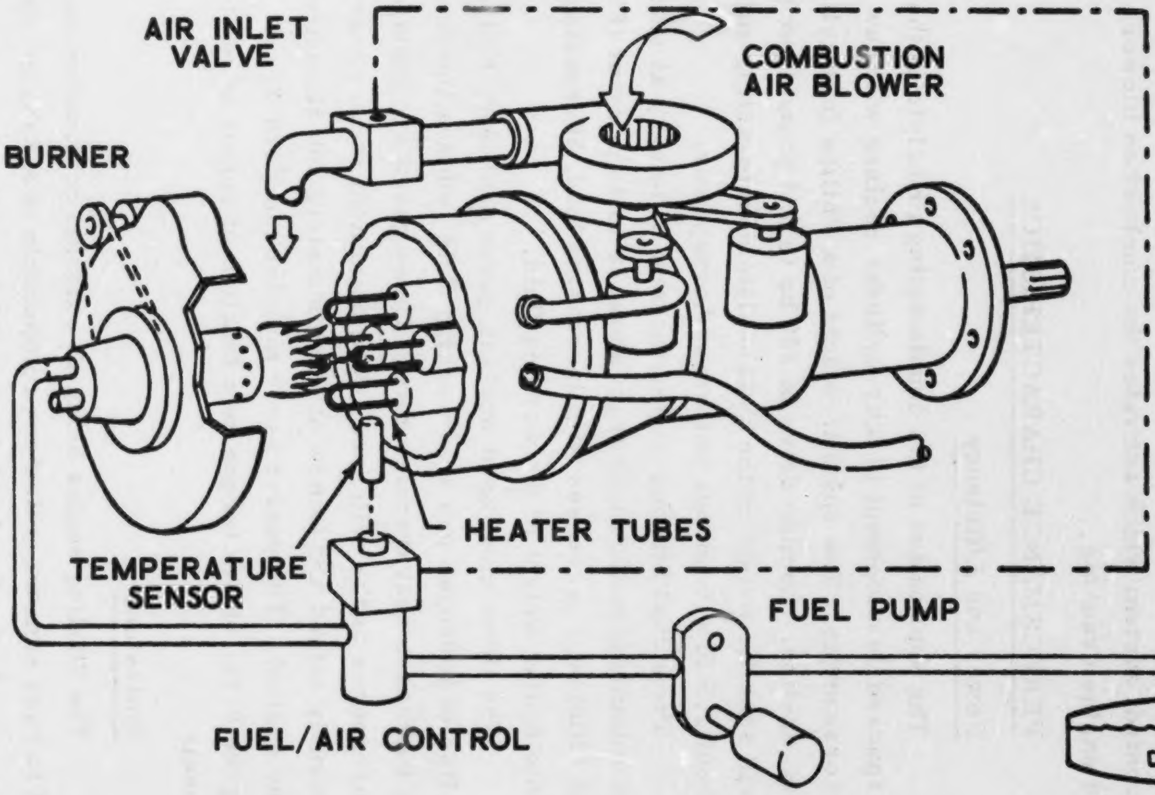

AIR INLET
VALVE
COMBUSTION
AIR BLOWER
BURNER
HEATER TUBES
TEMPERATURE
SENSOR
FUEL PUMP
FUEL/AIR CONTROL

Figure 5-17. Fuel Control Diagram (Ref. 5-1)

variables. A startup period of 10 to 15 seconds can be achieved (Ref. 5-5); however, necessary design compromises may require a period of up to 30 seconds. Starting characteristics are defined by the component parts of a sequential control system which activates the combustion blower, fuel flow, ignition, and engine cranking.

5.3 PERFORMANCE CHARACTERISTICS

5.3.1 Power and Efficiency

The application of the double-acting principle to the Stirling engine has spurred development of four-cylinder engines with low values of weight per horsepower. The specific weight of a Philips four-cylinder, single-action, in-line, rhombic drive of 200 hp is 8.8 pounds per horsepower. The projected specific weight of the swash-plate,double-acting automotive engine is about 3.5 to 4.5 pounds per brake horsepower.

The actual Stirling engine thermal efficiency at steady-state maximum to minimum load ranges from about 28 to 48 percent (Figure 5-4). The thermal efficiency decreases with increasing load, increasing rpm and increasing molecular weight of the working fluid.

The three most used working gases are: air, helium, and hydrogen. Use of hydrogen gas as a working fluid reduces internal-flow friction and increases heat transfer rate. These gases are installed at high mean pressures in the cylinders. The maximum mean pressure has been continuously raised and is now at approximately 200 atmospheres (Ford Torino engine). The performance map for the Ford Torino engine, given in Figure 5-18, shows torque as a function of engine speed for several pressure levels.

5.3.2 Emissions

The Stirling engine being an external combustion engine can be expected to have exhaust emissions comparable to a Rankine cycle engine. Very low mass emission levels of CO and HC have been reported for Stirling engines. While NO_x emissions are lower than for comparable uncontrolled-

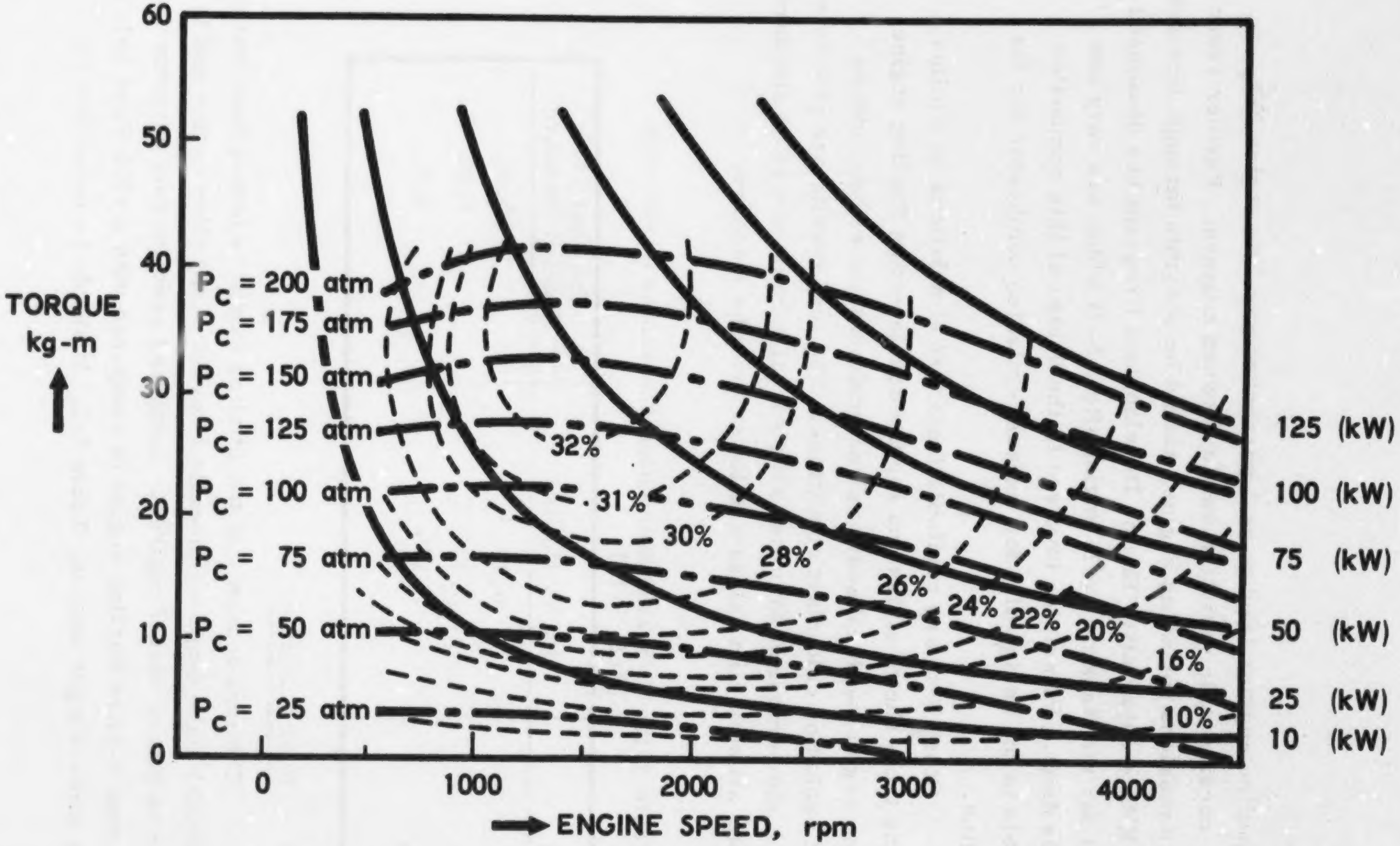


Figure 5-18. Performance Map of Ford Torino Stirling Engines (Ref. 5-1)

emission gasoline engines (as low as 1 percent of gasoline engine NO_x emissions), considerable variation exists between engines. Further reduction of NO_x emissions for some engines should be possible through directed research. EPA's Alternative Engine Development Program has developed a combustor for the Rankine cycle engine (Ref. 5-7) which has very low exhaust emissions. Principles involved in the design of this combustor are applicable to the design of a low exhaust emission combustor for the Stirling engine.

Other data are available from test simulations by Philips. The following table shows emissions for the double acting Stirling engine (Ford Torino engine) with exhaust gas recirculation but without add-on devices and excluding cold start conditions. The test results are presented in Table 5-1 and compared with the 1976 Federal exhaust emission standards. As indicated, all emission values are lower than the standard.

Table 5-1. CVS Test Simulation Emissions in gm/mile (Ref. 5-10)

Gas	N. V. Philips	Original 1976 Federal Standard
HC	0.10	0.41
CO	0.31	3.40
NO_x	0.175	0.4

5.3.3 Fuel Economy

The fuel economy of the Stirling engine is better than that of the conventional high-compression-ratio internal combustion engine and is comparable to that of diesel engines. Computed gasoline fuel economy for the 175-hp swash-plate Stirling engine is compared with a 1976 Ford 351, V-8 engine in a same-weight vehicle, Table 5-2. It should be noted that the overall efficiency, and thus fuel economy, of the Stirling engine is particularly dependent upon the coolant temperature. The calculated fuel economy given in this table assumed a coolant temperature of 163°F. A higher temperature resulting from an inadequate radiator or high ambient temperature would degrade the Stirling engine efficiency and, hence, its fuel economy.

Table 5-2. Fuel Economy Comparisons (Ref. 5-1)

	1976 Ford V-8	Stirling-Torino (Projected)	Gain
Consumer Average	10.7 mpg	14.7 mpg	37%
Suburban Driving	13.3 mpg	17.9 mpg	35%
City Traffic	8.2 mpg	11.4 mpg	39%
Superhighway	13.0 mpg	16.6 mpg	28%

5.3.4 Noise Levels

The Stirling engine noise level is about 10 to 15 dB(A) lower than for current passenger cars and about 15 to 20 dB(A) less than a normally silenced diesel engine. The noise level of a Philips Stirling engine installed in a bus was measured to be 68 dB(A). The largest noise sources are the combustion air blower and radiator fan. The absence of poppet valves, intermittent combustion noise, and high cylinder-pressure gradients contribute to its quiet operation. The noise level of a double-displacement Stirling engine is lower than that of the rhombic-drive engine because of the absence of synchronizing gears, which are the major noise source in a displacer engine.

5.3.5 Odor and Particulates

The Stirling engine exhaust odor is undetectable at concentrations normally encountered in the atmosphere. Smoke intensity is almost zero when the air-to-fuel ratio is greater than 26 to 1. Application of exhaust gas recirculation (EGR) reduces the NO_x emission and the CO and soot emissions at low loads. Increasing EGR at low loads does not detract powerwise, since the blower consumes little power at this condition.

5.3.6 Maintainability and Reliability

The Stirling engine's heat tubes, anti-leak seals and control system are the critical items with regard to maintainability and reliability. (A unique roll-sock seal has been developed by Philips to separate the working fluid from the lubrication system. Durability statistics are not completed for this device at this time.) These items can be investigated most meaningfully by operating a Stirling engine in a vehicle over the Federal Driving Cycle for at least 50,000 miles. At present, only laboratory data are available and do not represent environmental conditions as they are found in road operations.

5.3.7 Safety

A vehicle with a Stirling engine should be about as safe to operate as a vehicle equipped with a diesel engine. Both engines inject fuel into the combustion chamber, although the Stirling engine utilizes lower injection pressures. If the working fluid is hydrogen gas, some danger exists if the fluid escapes and collects in an enclosure where an explosion could occur. However, hydrogen gas diffuses rather rapidly; moreover, only a small quantity of gas is contained in the engine. If helium gas were used as the working fluid, the Stirling engine would be safer to operate than a diesel engine.

Toxicity is not present in the fluids currently considered for use in a Stirling engine; hence, it is not considered a safety problem.

5.3.8 Driveability

The Stirling engine has torque and speed characteristics comparable to current spark ignition engines (Ref. 5-5). Therefore, driveability of the vehicle should be similar to that of a vehicle powered by the spark ignition engine. Table 5-3 shows performance projections for the Ford Torino powered by a Stirling engine and another powered by a spark ignition engine. As shown, performance of the two engines is comparable.

Table 5-3. Performance Projections (Ref. 5-5)

	Stirling 4-215 Engine	354-2VC Passenger-Car Engine
Distance, 0-10 Sec. ft	491.0	484.0
Time, 0-60 Mph, sec	11.1	11.1
Time, 25-60 Mph, sec	8.0	8.0
Time 50-80 Mph, sec	11.2	11.3

5.4 CURRENT STATUS OF TECHNOLOGY

5.4.1 Current Utilization

The major research work on the Stirling engine was accomplished by Philips Research Laboratories of Holland. Philips has established that a double-acting engine with a swash plate is suitable for installation in a contemporary medium-size automobile.

A combined development program conducted by Ford and Philips has resulted in the following technology status for the Stirling engine (Ref. 5-1):

a. The Stirling engine has potential for low-emissions operation in a current domestic automobile.

b. The engine has the potential for very low noise level of operation.

c. The Stirling engine performance is similar to that of a spark ignition engine.

d. The Stirling engine has the potential for excellent fuel economy.

e. The Stirling engine can be packaged within the present-day engine compartment.

Ford, in addition to work with Philips, has an agreement with United Stirling of Sweden to develop a small V-4 Stirling engine for powering a Pinto car.

5.4.2 Current Research and Development

Current research and development activity is directed at proving the potential of the Stirling engine as an automobile power plant. Major open issues in Stirling engine development are:

a. Cooling system requirements
b. Use of hydrogen as a working fluid and for seal durability
c. Maintenance to ensure low emissions for 50,000 miles
d. Mass-production manufacturing cost.

Ford and Philips will concentrate on solving these problems and expect to install a prototype engine in a 1975 Ford Torino for performance verification road tests (Ref. 5-1).

5.5 PROJECTED STATUS

5.5.1 Potential for National or Regional Transportation

The Stirling engine was developed initially for special applications where performance was of utmost importance and manufacturing cost considerations were secondary. As stated in Ref. 5-11, the Stirling engine could, technically, replace the spark ignition engine; however, mass production of this engine will be delayed until economic feasibility can be proven. The engine cannot compete with the cost of the present day spark ignition engine.

Two scenarios could lead to increased emphasis for this engine as a candidate for future automobiles:

a. The cost of fuel increases to a point where the improved fuel economy of the Stirling engine offsets the higher first cost and its lifetime cost results in an overall economic advantage over the spark ignition engine.

b. Air pollution legislation dictates the use of a low emission engine like the Stirling engine.

In any event, even at an accelerated rate of research, it will be mid-1980 before this engine can become available for limited production.

5.5.2 Impact on Energy Requirements and Air Quality

As previously discussed, this engine is highly efficient and provides low emission levels. Therefore, it is an ideal engine from the viewpoint of combining conservation of energy with improvement in air quality.

6. DIESEL ENGINES

6.1 INTRODUCTION

6.1.1 General Description

The diesel or compression ignition engine is a reciprocating engine which is, in many respects very similar to the Otto cycle spark ignition engine. In the diesel, air alone is compressed in the cylinder, and the fuel is then injected into the heated air towards the end of the compression stroke. The temperature developed during compression is sufficiently high to ignite the fuel immediately after injection. Thus, spark plugs, distributor, and carburetor are eliminated in the diesel, but a high-pressure, fuel-injection system is required. Since compression-ignition demands high pressure ratios, the current diesel engines are heavier, more bulky, and costlier than equivalent spark ignition engines.

The principal advantages of diesels relative to spark ignition engines include their ruggedness and their higher thermal efficiency, which results in lower fuel consumption, particularly at part-load operating conditions.

Many different engine configurations have been built around the basic thermodynamic diesel cycle, and some of these are currently being marketed by many companies throughout the world. The various designs can be grouped into two categories -- open chamber and divided chamber -- and each of these can then be subdivided into different engine classes, depending upon specific features related to the design of the combustion chamber and the induction system. Open-chamber designs are favored by most manufacturers of heavy-duty diesels because of the better fuel economy obtained with this engine type. Conversely, divided-chamber designs (prechambers and swirl chambers) are preferred by the manufacturers of light-duty engines because of their higher speed capability.

6.1.2 Historical Background

The diesel engine was invented by Rudolf Diesel in 1892 and was originally intended to burn coal dust. In his early designs, Diesel proposed to coordinate the rate of fuel injection with the movement of the pistons in such a manner that the heat of combustion would be liberated at constant temperature, thus approaching the thermal efficiency of the Carnot cycle. However, this approach proved to be unsuccessful, and the cycle was then modified to achieve near-constant pressure combustion. While this ideal process is still being utilized in large, low-speed diesels, the high-speed diesels used in light-duty applications employ a process between the ideal cycle and the Otto cycle.

The high compression ratio required to achieve fuel ignition, and the resulting bulk, precluded the application of diesels in early road vehicles. The first diesel-powered truck was built in 1907 by Saurer in Switzerland, in cooperation with Diesel; in 1911 the first fully loaded diesel truck crossed the United States from coast to coast.

Daimler-Benz of Germany started series production of its Mercedes Benz 260D diesel-powered automobile in 1936. Since resumption of production in 1949, Daimler-Benz has sold more than one million diesel automobiles. A number of other companies have since entered the light-duty diesel engine field, including Peugeot of France, Perkins of England, Daihatsu of Japan, and Opel of Germany.

6.2 POWER PLANT DESCRIPTION

6.2.1 Diesel Cycle

The thermodynamic process describing the operation of the ideal diesel, or compression-ignition engine, is depicted in Figure 6-1 in terms of the cylinder pressure versus cylinder volume and the temperature of the combustion products versus system entropy. In the ideal cycle, the compression and expansion processes are isentropic processes; combustion is a constant-pressure process, and the blowdown of the exhaust gases is a constant-volume process.

In the ideal diesel cycle, air is inducted into the cylinder during the intake stroke of the piston and compressed isentropically during process ① - ② (Figure 6-1). Fuel injection into hot compressed air and ignition occur during process ② - ③. Injection and ignition take place in such a manner that the descending piston compensates for the pressure buildup that would otherwise occur as a result of the heat released during combustion. The combustion process is completed at point ③ and the high pressure gases then expand isentropically to point ④, thereby performing additional work on the piston. The exhaust valve opens at point ④ , and the bulk of the exhaust gases discharge at nearly constant volume. The remaining exhaust gases are then pushed out by the ascending piston.

The indicated thermal efficiency of the ideal diesel cycle is presented in Figure 6-2 for two specific heat ratios, k = 1.3 and 1.4. Also shown for comparison are curves representing the ideal Otto cycle. For a given compression ratio and specific heat ratio, the indicated thermal efficiency of the diesel cycle is lower than that of the Otto cycle. However, diesels are normally designed to operate at much higher compression ratios (of the order of 15:1 to 20:1) than spark ignition engines (typically 9:1), and this improves the efficiency of the diesel with respect to the spark ignition engine. Also, the specific heat ratio of the diesel combustion products is higher than for spark ignition engines, especially under part-load conditions which further improves the diesel cycle efficiency. These effects are responsible for the higher efficiency of diesel engines relative to gasoline engines.

In actual diesels, the combustion process is modified somewhat from the ideal process illustrated in Figure 6-1. In the real engine, a substantial portion of the total fuel is injected slightly before top dead center (TDC), and combustion of that fuel occurs at essentially constant volume, with the piston at TDC. Combustion of the remainder of the fuel takes place at nearly constant pressure while the piston starts to descend from TDC. The indicated thermal efficiency of this "dual cycle" process falls between the diesel cycle and the Otto cycle.

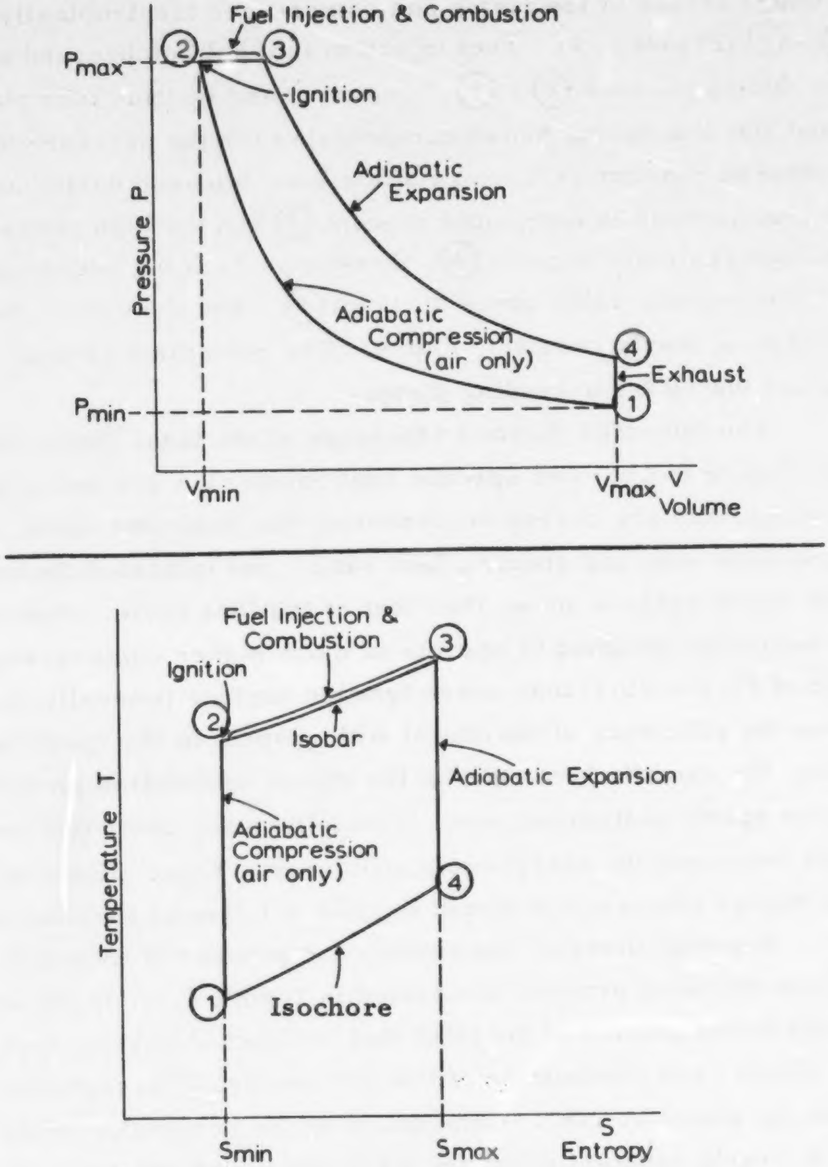

Fuel Injection & Combustion
2
3
Pmax
Ignition
Adiabatic Expansion
Pressure P
Adiabatic Compression (air only)
4
Exhaust
1
Pmin
Vmin
Vmax
V
Volume
Fuel Injection & Combustion
3
Ignition
2
Isobar
Adiabatic Expansion
Temperature T
Adiabatic Compression (air only)
4
1
Isochore
S
Smin
Smax
Entropy

Figure 6-1. Ideal Diesel Cycle

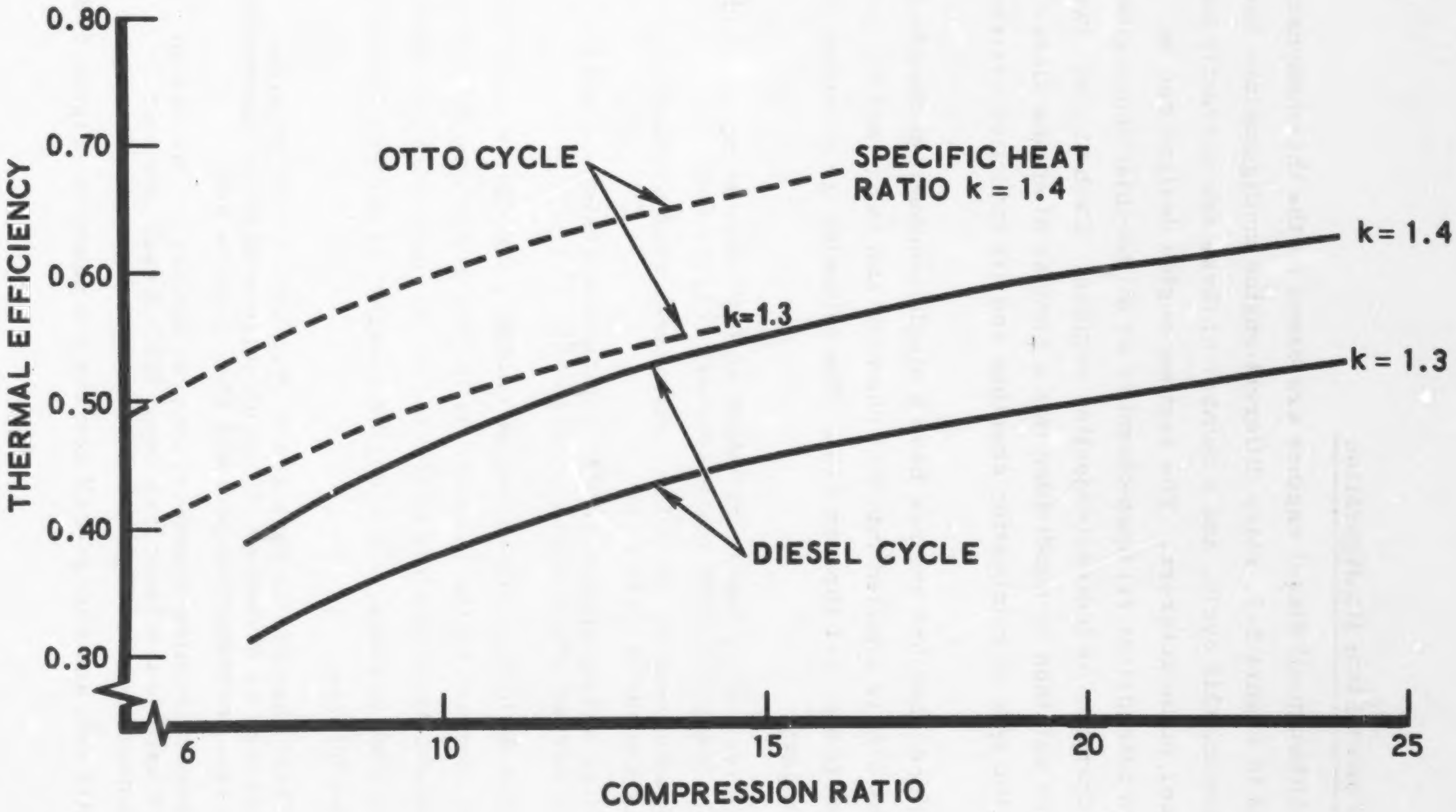


Figure 6-2 . Diesel and Otto Cycle Thermal Efficiency vs Compression Ratio

6.2.2 Power Plant Configuration

Although all diesel engines are based on the thermodynamic cycle illustrated in Figure 6-1, many different engine configurations have been designed around this cycle, and a number of these are currently being marketed by many manufacturers. The various engine designs can be grouped into two categories: (1) open-chamber or direct-injection engines, and (2) divided chamber or indirect-injection engines. Each of these two engine categories can then be subdivided into a number of engine classes, depending upon the type of combustion chamber and air induction system utilized.

Open-chamber engines have a single combustion chamber per cylinder, and the fuel is injected into this chamber and vaporized for subsequent combustion (Refs. 6-1 through 6-3). The following three engine classes fall into this category:

a. Quiescent or low-swirl "Mexican hat" chambers, favored by many domestic manufacturers (Figure 6-3)

b. Medium-swirl, deep-bowl chambers, used primarily in Europe (Figure 6-4)

c. High-swirl, spherical "M" combustion chambers used in special applications (Figure 6-5).

The divided-chamber engine category employs a main combustion chamber, formed by the cylinder walls, the piston and the cylinder head, and an antechamber which is connected to the main chamber through a communicating flow passage. This engine category is further divided into three classes, as follows:

a. Swirl chamber or turbulence chamber, a configuration favored by a number of manufacturers of diesel passenger cars, including Peugeot and Opel (Figure 6-6)

b. Precombustion chamber, or prechamber. This design is used in the Mercedes Benz 220D diesel-powered automomobile (Figure 6-7)

c. Air cell and energy-cell combustion chamber (Figure 6-8)

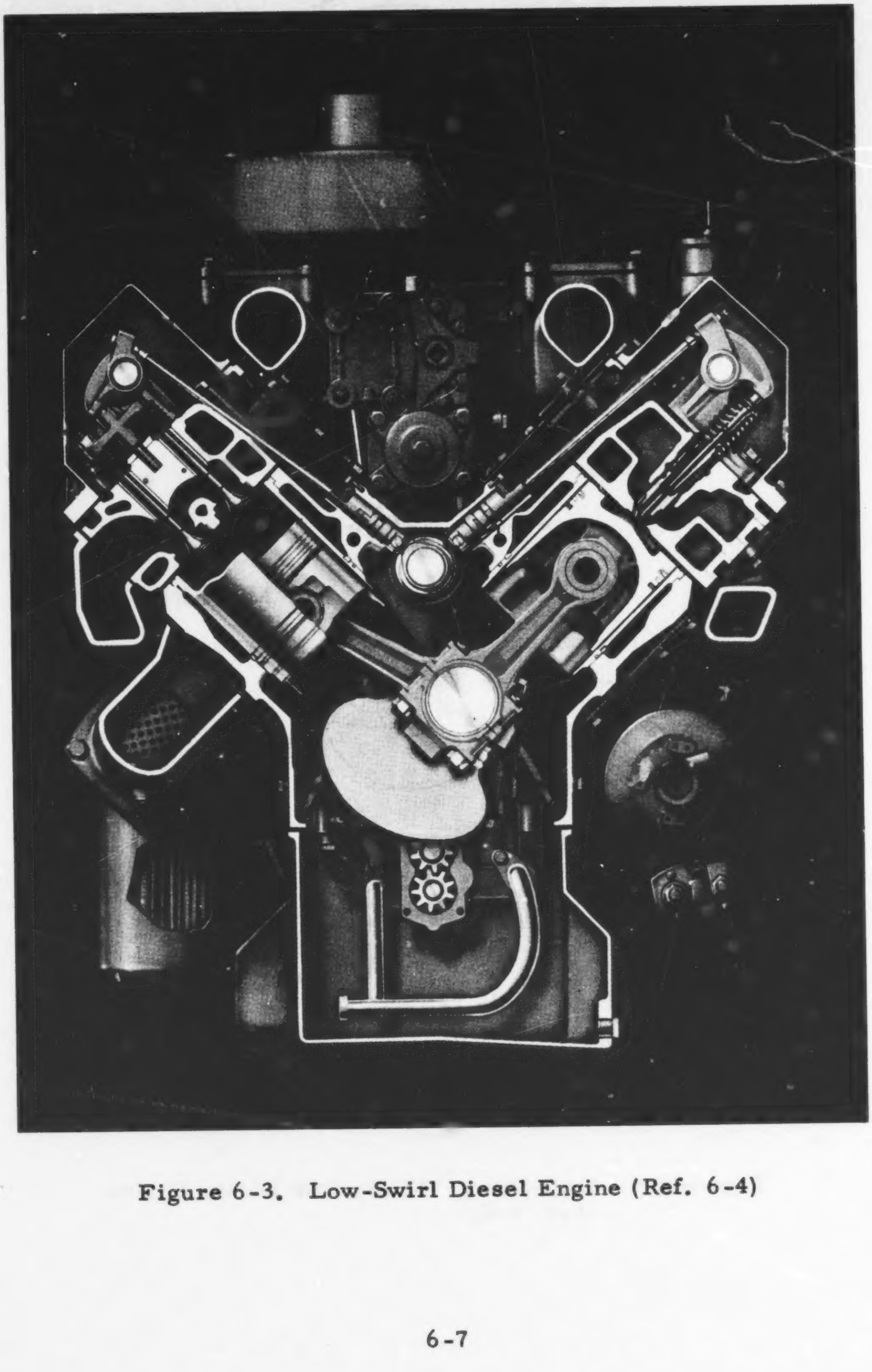


Figure 6-3. Low-Swirl Diesel Engine (Ref. 6-4)

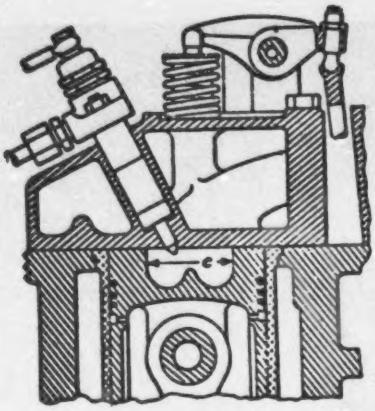


Figure 6-4. Medium-Swirl Diesel Engine (Ref. 6-5)

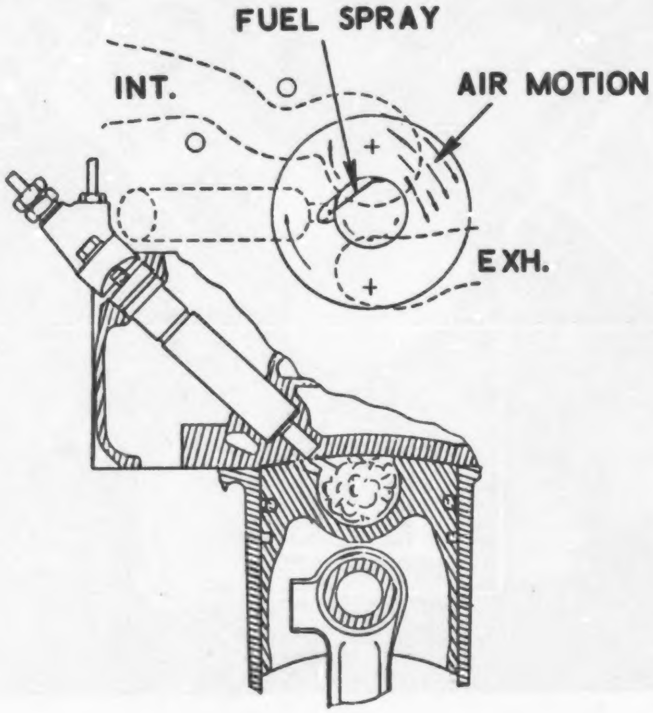

FUEL SPRAY
INT.
AIR MOTION
EXH.

Figure 6-5. High-Swirl, "M" System Diesel Engine (Ref. 6-6)

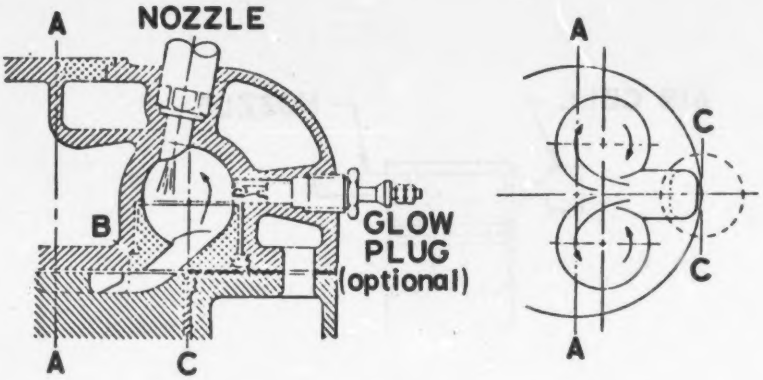

A
NOZZLE
A
C
GLOW
PLUG
(optional)
B
C
A
A
C

Figure 6-6. Ricardo Swirl Chamber Diesel (Ref. 6-7)

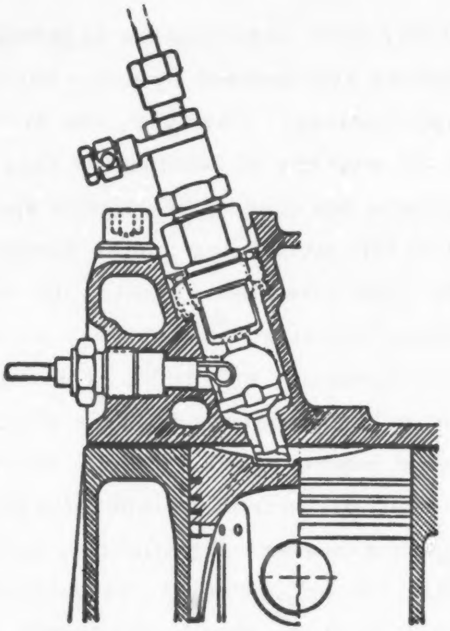


Figure 6-7. Daimler-Benz Precombustion Chamber Design (Ref. 6-8)

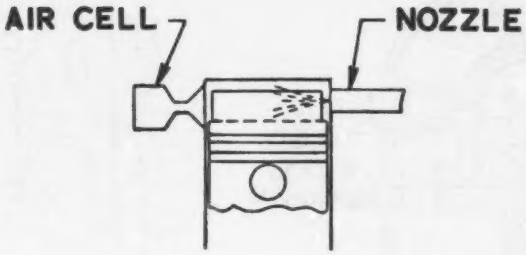


Figure 6-8. Air-Cell Diesel Engine Schematic (Ref. 6-1)

Because of lower fuel consumption (approximately 5 to 8 percent), direct injection engines are favored by many manufacturers for use in industrial and truck applications. However, the divided-chamber configurations are preferred by the makers of automotive diesels and by some of the truck engine manufacturers because of the higher speed capability and the lower emission potential of this particular engine design.

Compared with open-chamber diesels, the divided-chamber engine has a number of additional inherent advantages, as well as disadvantages. Because of its higher volumetric efficiency, the engine can be operrated at higher brake mean effective pressures. This improvement is realized by employing near-zero intake swirl. Furthermore, at a given compression ratio, the maximum pressures, temperatures, and pressure-rise rates in divided-chamber engines are significantly lower than for open-chamber designs. As a result, the engine noise, exhaust emissions, and mechanical stress characteristics of the engine are improved accordingly.

The principal disadvantages of divided-chamber diesels compared with open-chamber configurations include higher fuel consumption and higher exhaust gas temperatures. The loss in fuel economy is primarily due to the higher surface-to-volume ratio of the engine and, in the case of the swirl chamber configuration, the high swirl velocities in the combustion chamber.

Daimler-Benz has chosen the low-swirl prechamber design for its 220D passenger car for a number of reasons, including the high-speed capability of the engine and its high volumetric efficiency and low noise and emission levels. To enhance ignition at idle and low loads, Mercedes utilizes a ball pin located in the center of the prechamber. Upon ignition, this pin heats up very rapidly and maintains a temperature of about 1300°F under all operating conditions. The fuel injected into the chamber impinges on the ball pin, thereby providing a stable ignition source and assuring brief ignition delays at all times. During cold start of the engine at ambient temperatures below about 70°F, activation of the electrically heated glow plug is required for a few seconds in order to minimize engine cranking.

6.2.3 Design Features

6.2.3.1 Power Output

The specific power output of heavy-duty and light-duty diesels in terms of maximum horsepower per cubic inch displacement is presented in Figure 6-9 as a function of maximum horsepower. Also shown are correlations established for domestic and imported 1971 model year light-duty gasoline engines (Ref. 6-9). The specific power output of the smaller spark ignition engine used in imported cars is about 40 to 50 percent higher than for domestic passenger car engines. This trend is the direct result of the higher maximum speed of the imported engines. As indicated in the figure, the specific power output of the passenger car diesel engines manufactured by Daimler-Benz and Opel is only slightly lower than that of 1971 domestic

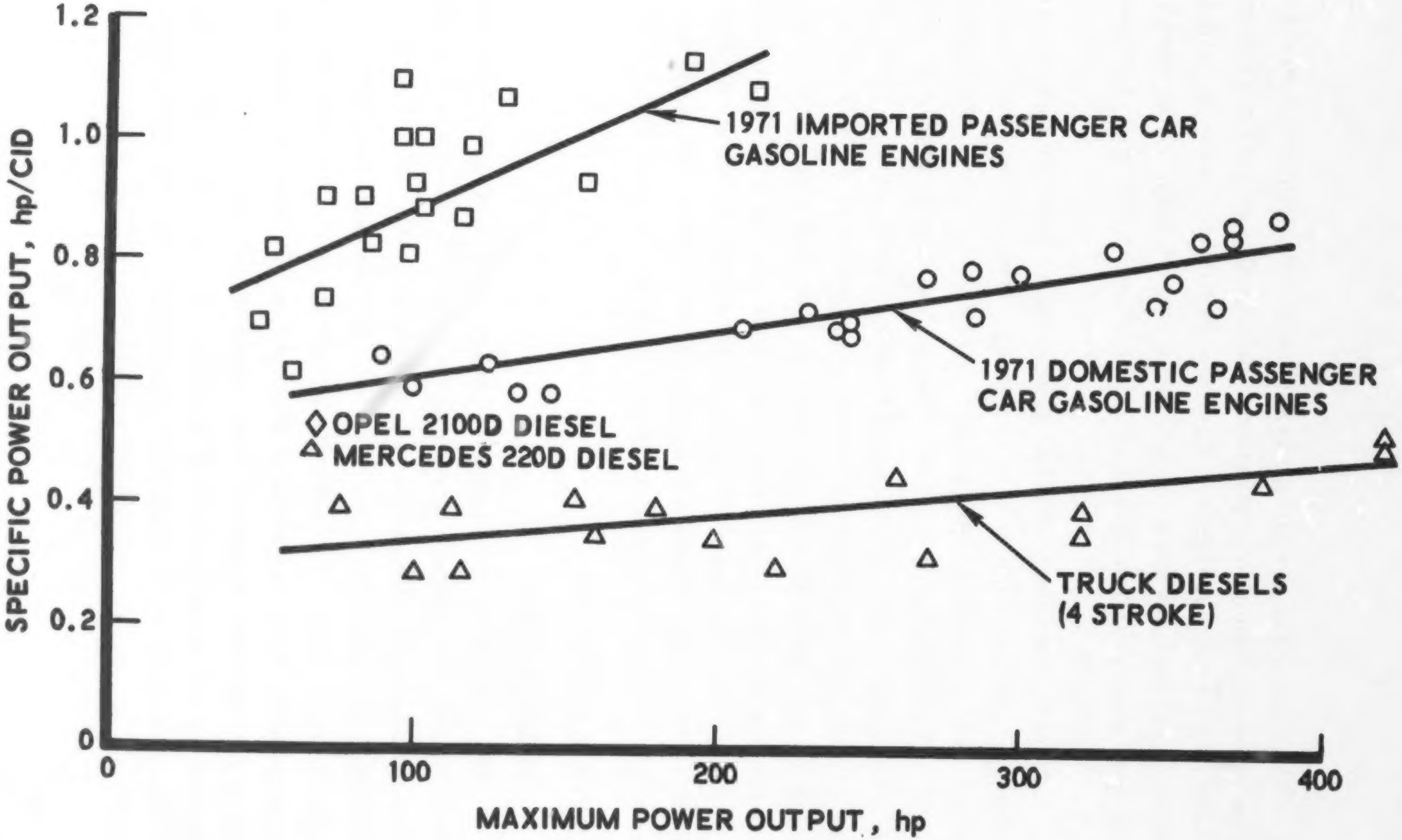


Figure 6-9. Specific Power Output of Diesel and Gasoline Engines (Maximum Horsepower/Cubic Inch Displacement)

spark ignition automobile engines, but only about 60 to 70 percent of the specific power of the smaller, higher-speed engines used in imported vehicles.

A number of factors contribute to the relatively low specific peak-power output of current diesels, in spite of the higher thermal efficiency of the diesel cycle. These include (1) the lower maximum speed capability of the diesel which is limited by the stress loading of the pistons, connecting rod, crankshaft, and bearings, (2) the lean air-fuel ratio operation, and (3) the loss in volumetric efficiency occurring at high engine speeds (Ref. 6-12). In divided-chamber diesels, additional power losses are due to the pressure drop occurring in the flow passage connecting the two chambers. However, these losses are partially compensated for by the higher compression ratio generally used in divided-chamber engines.

The power output of diesel engines can be substantially increased by means of turbocharging or supercharging. As a result, the specific power output of the engine increases accordingly and can exceed that of emission-controlled gasoline engines. Turbochargers are frequently utilized in truck applications to increase the peak power output of the engine, especially at altitude, and to improve its fuel economy. However, in the low power regime of the engine, the turbocharger is generally much less effective, and its response to load variations of the type encountered in passenger cars is rather slow because of the inertia of the turbine and compressor rotors. These problems are alleviated to a large extent in the Comprex supercharger system, which has been developed by Brown Boveri of Switzerland for use in heavy-duty diesel engine applications (Refs. 6-10 and 6-11). This device, shown schematically in Figure 6-10, is a compression wave machine which utilizes the energy of the exhaust gases to compress the intake air before admission into the cylinder. Relative to turbochargers, the Comprex system has a number of advantages, including (1) fast response to engine load variations; (2) higher BMEP capability at low engine speeds; (3) "internal" EGR, which can be achieved by proper timing of the intake valve closing; and (4) lower smoke and NO_x emissions. The disadvantages

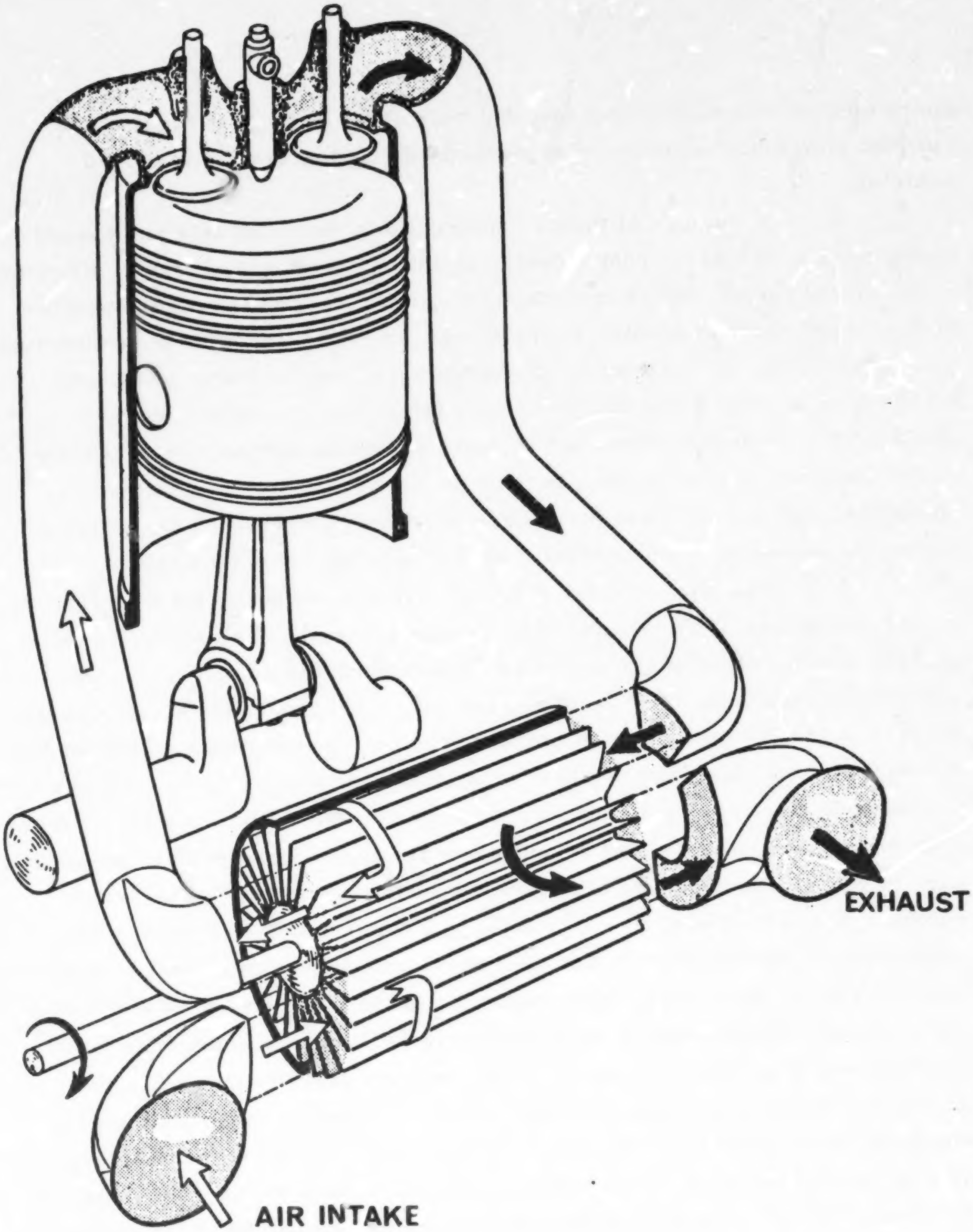

EXHAUST
AIR INTAKE

Figure 6-10. Comprex Supercharger Arrangement
(Ref. 6-10)

of the Comprex system include its larger size and higher weight, cost, and noise levels.

6.2.3.2 Specific Weight

Specific weight data (pounds per horsepower output) for current automotive and industrial open-chamber and prechamber diesels are plotted in Figure 6-11. The open-chamber data were utilized to establish the trend of specific weight with rated horsepower. The correlation shown for divided-chamber engines was then obtained by transferring that slope to the mean level of the divided-chamber data. The two curves show that divided chamber engines are of the order of 25 percent lighter than equivalent open-chamber designs. This weight difference is directly related to the lower peak combustion pressures developed in divided chambers resulting in weight savings in the cylinder head, engine block, connecting rods, crankshaft, and bearings. Typically, the peak pressures in divided-chamber engines are between about 1,000 psi and 1,200 psi, compared with about 1,400 psi to 2,000 psi for open-chamber configurations. The peak combustion pressures in spark ignition engines are of the order of 600 psi for low compression ratio engines, and as high as 1,000 psi for high-compression-ratio designs.

The specific weight of automotive reciprocating spark ignition engines is also presented in Figure 6-11. This correlation is based on 25 engines ranging from 50 to 350 brake-horsepower output. The weights shown include the starter, alternator, and flywheel, but exclude the radiator and the cooling water (Ref. 6-14).

Comparison of the data indicates that current diesels are considerably heavier than equivalent gasoline engines. For example, a 100-horsepower, naturally-aspirated, divided-chamber diesel weighs about 620 pounds compared with only 350 pounds for the gasoline engine. With 40 percent turbocharging, the weight of this engine can be reduced to about 500 pounds.

Additional weight savings might be possible by (1) increasing the degree of turbocharging, (2) reducing the peak combustion pressures, and (3) lowering the design life of the diesel engine. Currently, the life of

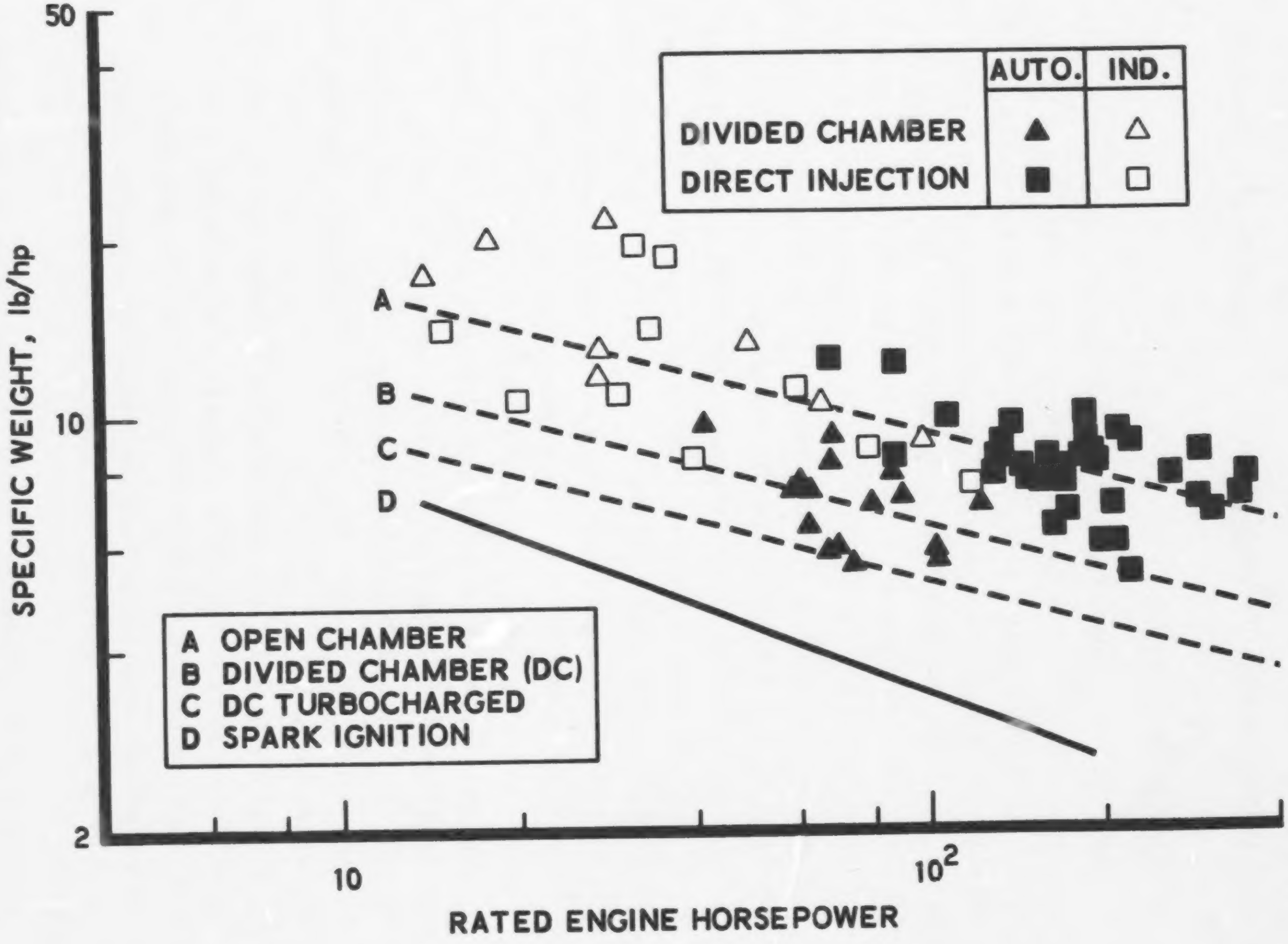


Figure 6-11. Specific Weight of Diesel and Spark Ignition Engines (Ref. 6-13)

truck diesels before major overhaul is of the order of 300,000 miles or more, compared with about 80,000 miles for light-duty gasoline engines. The specific weight of newly designed light-duty diesels is estimated to be comparable to that of gasoline engines (Ref. 6-15).

6.2.3.3 Specific Volume

The specific volume (ft^3/hp) of current heavy-duty diesel engines is shown in Figure 6-12 as a function of rated engine power. There is no reason to expect biases between open and divided chambers and no attempt was made to correlate the data on that basis. The turbocharged engine correlation shown Figure 6-12 was drawn on the basis of a 40 percent realized power increase, compared with naturally aspirated diesels. Also presented in Figure 6-12 is the specific volume curve for reciprocating spark ignition engines. Both curves are based on the total engine envelope from fan to flywheel and from air cleaner to oil pan.

As indicated Figure 6-12, current diesel engines are considerably larger than equivalent gasoline engines. For example, a 100-horsepower, naturally-aspirated diesel engine has a volume of about 17 ft^3 compared with about 10 ft^3 for the gasoline engine. However, the size of the diesel could be reduced substantially by turbocharging or supercharging.

6.2.3.4 Manufacturing and Materials

Basically, the fabrication techniques, tolerances, tooling, and materials used in the manufacture of automotive diesels are quite similar to those employed in gasoline engines. However, according to Daimler-Benz, some modifications and additions to the transfer lines of spark ignition engines are needed to account for the additional steps required in the manufacture of diesel engines (Ref. 6-12). Although the gasoline and diesel engines utilized by Mercedes in its 220 and 220D vehicles were originally designed to use common parts whenever possible, there are many specific design differences. These include the cylinder head (which is cast iron in current diesels and aluminum in the spark ignition engine), engine block, head gasket, pistons, rings, crankshaft, intake manifold, and starter. Similar materials

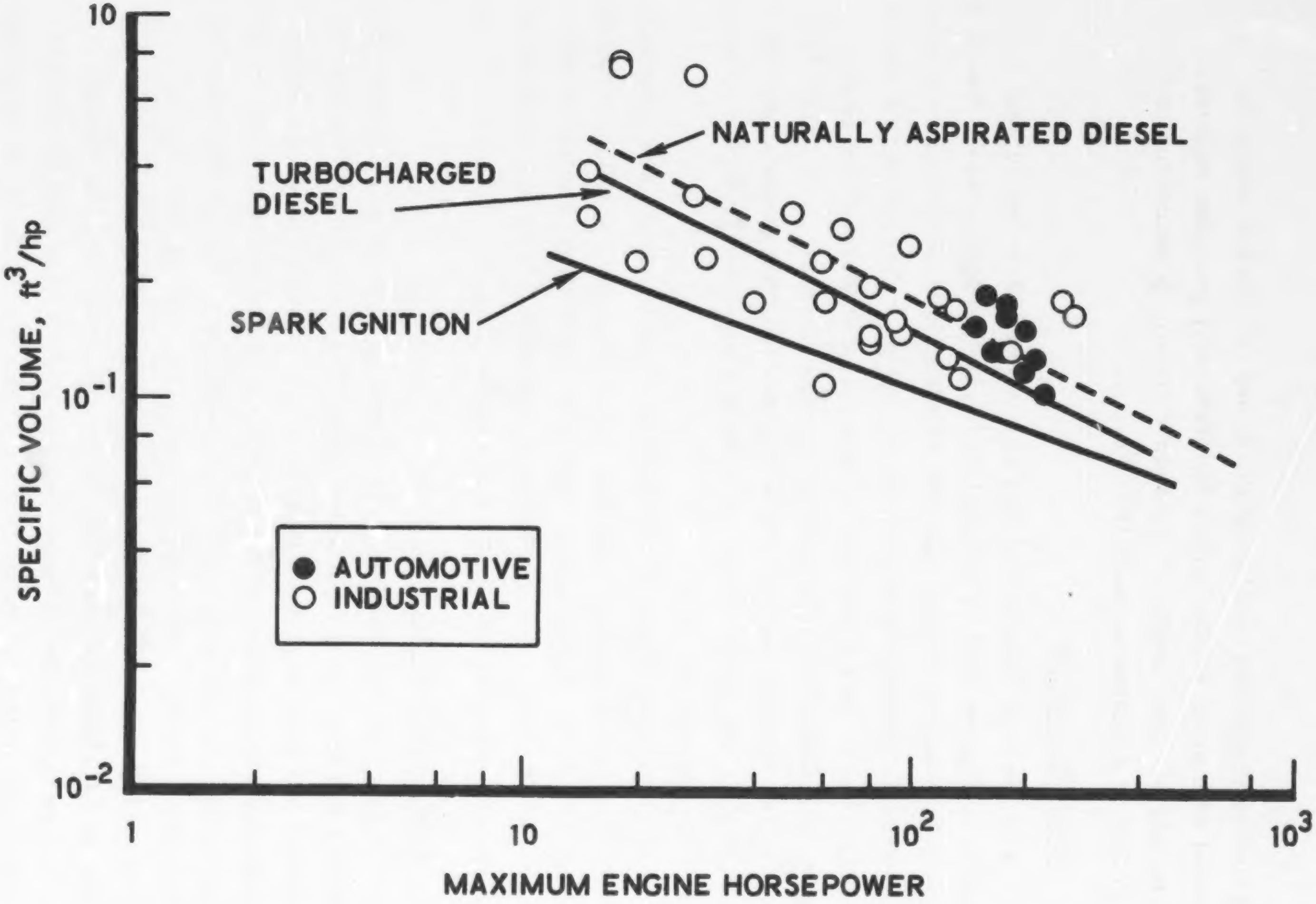


Figure 6-12. Specific Volume of Diesel and Spark Ignition Engines (Ref. 6-13)

and fabrication procedures are currently utilized by the other manufacturers of light-duty and heavy-duty diesel engines.

6.2.3.5 Fuel Requirements

Although most diesels run quite well on a variety of fuels, certain fuel properties are required to achieve optimum conditions in terms of engine performance and smoke emissions. Cetane number is by far the most important fuel parameter. For good combustion, the cetane number, which is a measure of the autoignition characteristics of the fuel, should be sufficiently high to minimize the ignition delay time. Cetane numbers and autoignition temperatures of a number of fuels are listed in Table 6-1. Experience has indicated that little improvement in the combustion process is obtained for cetane numbers above about 50 or 55 (Ref. 6-17).

As indicated in Table 6-1, the autoignition temperature of low-cetane fuels is high, making ignition more difficult to achieve. Therefore, diesel engines designed to operate with these fuels generally have a higher compression ratio to ensure ignition and stable combustion during starting and at low power conditions. Since the occurrence of ignition in diesels depends on the heat transfer into the fuel droplets, the maximum feasible engine speed decreases with decreasing cetane number. Preheating the air charge counteracts that problem to a large degree.

The sulfur in the fuel has a significant effect on the life of the cylinder liner, piston rings, and bearings of reciprocating engines. When using high-sulfur fuels, frequent oil changes are desirable to counteract the problems related to acid formation. For example, if the sulfur content increases about 0.5 percent, Daimler-Benz reduces the oil change interval for its light-duty automotive diesel engine from 3,000 miles to 1,500 miles.

With respect to fuel effects on diesel exhaust emissions, a considerable amount of work has been conducted by several investigators. Data published by Marshall and Hurn (Ref. 6-18) indicate very little change in the hydrocarbon, carbon monoxide, and oxides of nitrogen emissions of several diesel engines when operated on No. 1 and No. 2 diesel fuel and on two different solvents. Similar results were obtained by Daimler-Benz for

Table 6-1. Cetane Numbers and Autoignition Temperatures of Different Fuels (Ref. 6-16)

Fuel	Cetane Number	t_i, F Ignition Temp. at 450 psi
Ether	100	330
Cetane ($C_{16}H_{34}$)	100	380
n-Heptane (C_7H_{16})	60	380
Diesel Fuel A	55	450
Diesel Fuel B	45	480
Lubricating Oil, 30 SAE	45	460
Kerosene	37	550
Gasoline		
Low Octane	20	570
High Octane	10	750
Methanol	10	720
Ethanol	10	640
Benzene	3	840
α-Methylnaphthalene ($C_{11}H_{10}$)	0	850

diesel fuels refined both in Europe and in the United States (Ref. 6-12). Conversely, tests conducted by Shahed, et al. (Ref. 6-19) on a four-stroke, single-cylinder engine (4.75-inch bore, 4.75-inch stroke, compression ratio 14.2 : 1) using three different fuels with cetane ratings of 40, 52, and 56 indicate some reduction in NO_x and specific fuel consumption as cetane number is increased.

6.2.3.6 Fuel Injection System

In diesel engines, the fuel injection system assumes the function of the carburetor used in gasoline engines. It is probably the most important component of the engine with respect to combustion performance, emissions, and noise generation, especially in large, open-chamber engines. With increasing swirl rates and air turbulence, the effects of the fuel injection system characteristics on combustion become less pronounced.

The fuel injection system consists of a high-pressure fuel pump, a fuel distributor, and a series of injection nozzles. Typical injection pressures are in the range of 2,000 psi to 30,000 psi. The system is designed to provide a high degree of fuel atomization and distribution within a sharply defined injection period. As further discussed in Section 6.3.3, injection timing has a very strong effect on NO_x and smoke emissions.

In general, fuel injection systems have the capability to supply a precisely metered amount of highly atomized fuel to the engine cylinder at all times. Obviously, this has a very beneficial effect on fuel economy as well as on HC and CO emissions.

6.2.3.7 Cooling Requirements

The cooling system utilized in diesel engines is very similar to that of gasoline engines and normally consists of a radiator, a belt-driven cooling fan and water pump, and a thermostatic control device.

In divided-chamber diesels, the surface-to-volume ratio of the two combustion chambers is higher than in open-chamber engines, resulting in high thermal loading of the cylinder head and pistons. Although design solutions to the thermal loading problems are available for the cylinder head, the overall heat flux into the engine block must be handled by the cooling system (Ref. 6-20). In general, the radiator size required for divided chamber diesels is about 30 percent to 50 percent larger than for equivalent open-chamber diesels.

6.2.3.8 Lubrication System

The lubrication systems used in automotive diesel and gasoline engines are similar, except for an additional oil circuit required in diesels for lubrication of the fuel injection pump. Typically, the oil flows from the sump through a strainer and filter and then into the main passages of the engine block.

Diesel engines require more stringent lubricating oil specifications than gasoline engines, especially with respect to its detergent properties, and acid neutralization and antiwear characteristics. Also, adherence to the oil viscosity grade recommended by the manufacturer is more important in diesels than in gasoline engines because of a more rapid increase in the oil viscosity with mileage accumulation. This increase is directly related to the higher blowby normally obtained in diesels. Therefore, oil changes are recommended more frequently for diesels than for gasoline engines. For instance, the recommended intervals for oil and oil filter changes are 3,000 miles for the diesel-powered Mercedes 220D and Opel Rekord 2100D automobiles. Peugeot recommends an even shorter oil change interval of 1,500 miles.

6.2.3.9 Transmissions

Both manual and automatic transmissions are feasible for use in diesel-powered automobiles. The Daimler-Benz 220D and the Opel Rekord 2100D passenger cars are available with either transmission type, while the Peugeot 504 diesel is offered with a four-speed manual transmission only. The transmissions of these vehicles are similar in design and construction to those used in the corresponding gasoline-powered cars marketed by these companies.

The automatic transmission by Daimler-Benz consists of a four-speed planetary gear set and a hydraulic clutch. Gear selection is controlled by engine speed and accelerator pedal position. The gears of this transmission have been optimized with respect to engine torque and speed.

Opel's automatic transmission represents a modification of the basic unit used in its spark ignition engine powered Rekord models. The design changes include the torque converter casing, oil pump, and band servo spring. In the absence of an intake throttle in the Opel diesel, the vacuum required for the gearbox control system is provided by a separate vacuum pump.

6.2.4 Operating Characteristics

6.2.4.1 Power Control Requirements and Methods

With few exceptions, heavy-duty diesel engines operate without intake throttle, while light-duty engines generally utilize intake throttling. In the absence of an air-flow control valve, the power output of the diesel is controlled entirely by the amount of fuel injected into the cylinders. At idle, the engine operates at air fuel mixture ratios of the order of 150:1, and the idle fuel consumption is only about 25 percent to 40 percent of that of gasoline engines. With increasing power demand, the fuel flow rate is increased by moving the accelerator pedal which is directly connected to the fuel injection pump. At full power, the air fuel ratio of automotive diesels varies between 20:1 and 25:1, depending upon the smoke characteristics of the particular engine and the permissible smoke levels.

6.2.4.2 Starting Characteristics

In general, the starting characteristics of diesel engines, when warm, are quite comparable to spark ignition engines. However, difficulties can arise during cold starts, particularly at very low ambient temperatures. In divided-chamber diesels, the starting problem can be alleviated by using an electronically heated glow plug in each prechamber. The purpose of the glow plug is to heat the air in the prechamber and to provide a hot spot for initial ignition. According to Daimler-Benz (Ref. 6-12), the degree of "preglowing" needed before engaging the starter motor varies with ambient temperature and, more importantly, with the temperature of the engine. Typically, with a cold engine and an air temperature of 68°F, the required preglowing time is between 5 and 10 seconds. However, under extremely low engine and ambient temperatures, as much as one minute of preglowing might be necessary. Unlike gasoline engines, diesels are less likely to stall after a successful start.

To provide sufficient energy for starting, the battery used in the Mercedes 220D automobile is about 30 percent larger than that of the gasoline engine powered 220 vehicle. Also, the starter motor in the 220D is more powerful (2.4 hp vs 1.5 hp for the gasoline engine) to compensate for the higher compression ratio of the diesel.

The starting time of open-chamber and divided-chamber diesel engines can be reduced by incorporating certain starting aids, at the expense of added complexity and cost. These devices include intake flame heaters, ether injection, and preinjection into the cylinders or intake manifold (Ref 6-1). Other approaches that have been considered include removal of the air swirl during engine starting (Ref. 6-6) and incorporation of variable compression ratio pistons (Ref. 6-14).

6.3 PERFORMANCE CHARACTERISTICS

6.3.1 Power, Speed, Torque, Efficiency, Combustion

Representative performance curves for a light-duty automotive diesel engine, operated at maximum fuel flow rates, are depicted in Figure 6-13. The power, torque, and specific fuel consumption (SFC) curves shown were computed from data published by Daimler-Benz for its 2-liter displacement (121.3 CID) prechamber engine which was used in the Mercedes 200D automobile (Ref. 6-8). For comparison, wide-open throttle power and torque curves for the Mercedes 2-liter spark ignition engine (not modified for emission control) are also presented in Figure 6-13.

The curves clearly illustrate the lower power output capability of the nonsupercharged diesel engine. In the low speed regime, the torque of the Mercedes 200D diesel is only about 20 percent lower than that of the uncontrolled Mercedes 200 gasoline engine. This explains the rather good low-speed acceleration capability of the 200D vehicle compared with equivalent

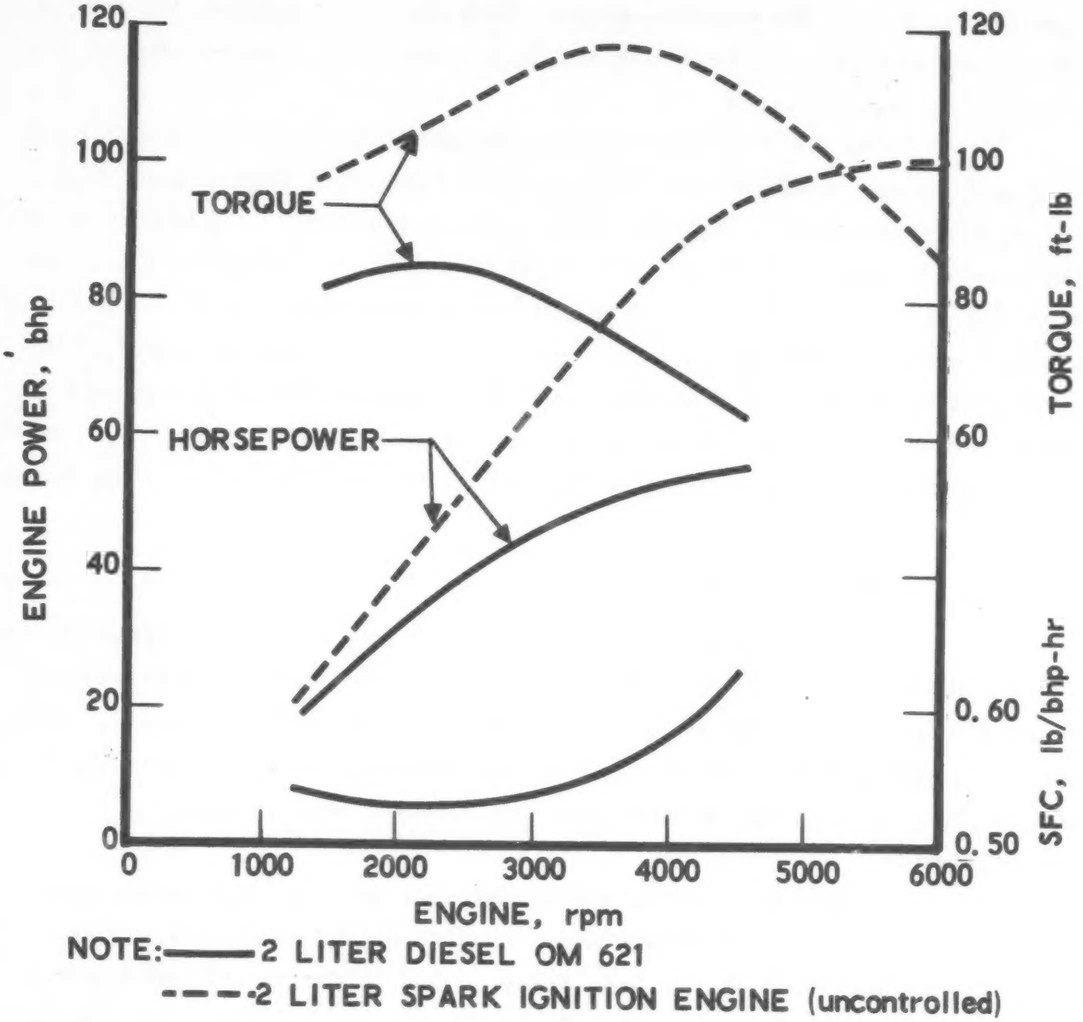


Figure 6-13. Diesel and Gasoline Engine Performance Parameters, Daimler-Benz Automobile Engines (Ref. 6-8)

gasoline engine powered cars. However, the torque decrement increases with increasing engine speed. For example, at the maximum speed of the 2-liter diesel (4,350 rpm), the torque of this engine is only slightly above 50 percent of that of the gasoline engine (Ref. 6-8). Of course, the torque of the diesel engine could be substantially increased by incorporating a turbocharger or supercharger.

The specific fuel consumption (SFC) of the Mercedes 2-liter diesel and uncontrolled gasoline engines is presented in Figure 6-14 as a function of engine speed and brake mean-effective pressure (BMEP). As indicated in the figure, minimum SFC of the diesel is only slightly lower than that of the gasoline engine. However, at low engine loads, the diesel exhibits superior fuel consumption characteristics. This is the direct result of the very lean air-fuel mixtures at part load, the lower pumping losses, and the higher compression ratio of the diesel. As discussed in Section 6.2.1, each of these factors has a beneficial effect on the thermal efficiency of the diesel cycle.

6.3.2 Diesel Combustion

The diesel combustion process is preceded by an ignition delay period, which occurs between the injection of the fuel into the compressed hot-air charge and the initiation of combustion (Refs. 6-21 and 6-22). This delay time, composed of physical delay and chemical delay, is dependent upon the fuel, the charge temperature, and certain engine design characteristics.

In prechamber and swirl chamber diesels, fuel injection is timed so that combustion commences near TDC for optimum performance. Under these conditions, the ignition delay period is short; and combustion proceeds smoothly and rapidly, but relatively late in the cycle. By the time combustion progresses to the main chamber, the piston is already descending. Therefore, conditions are changing rapidly toward reduced temperatures, thus minimizing the formation of nitrogen oxides (Ref. 6-23).

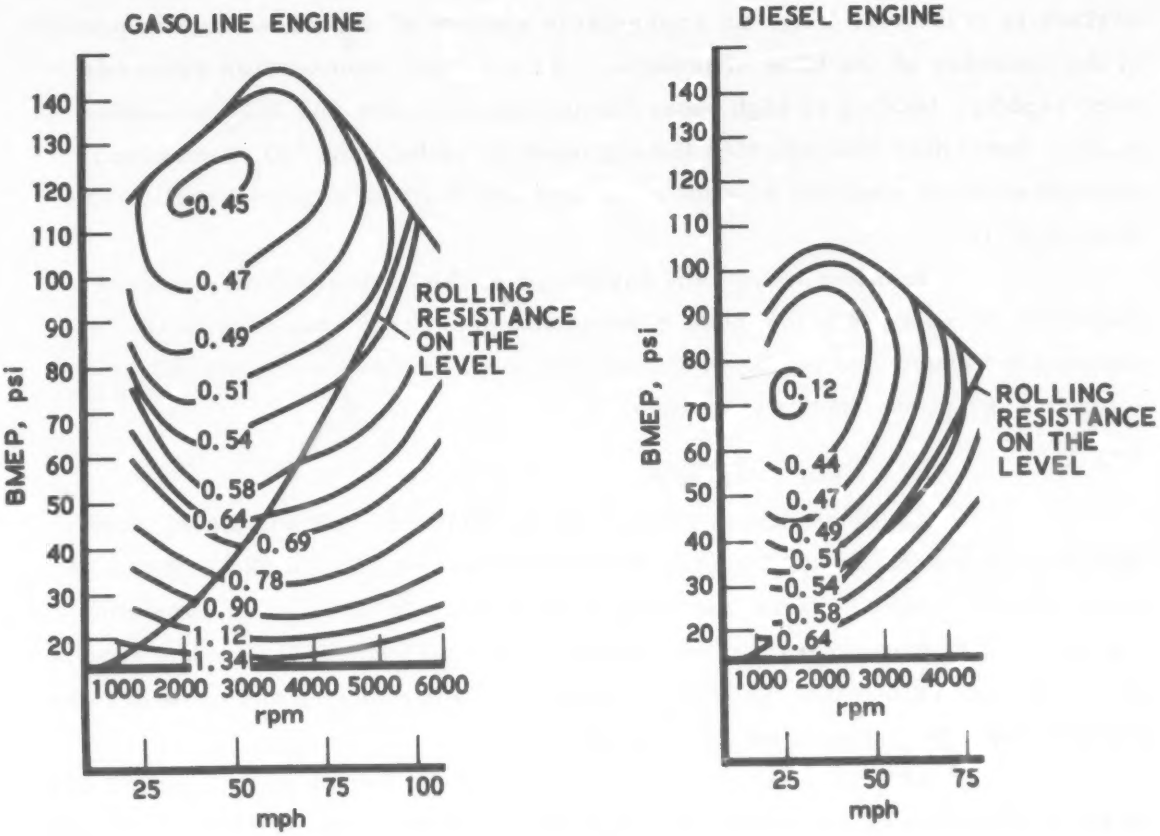


Figure 6-14. Precombustion Chamber Diesel and Uncontrolled Spark Ignition Engine Performance Maps (Ref. 6-8)

The combustion in open-chamber engines proceeds in a different manner. For optimum smoke-free performance, combustion commences prior to reaching TDC. Since the ignition delay period in these engines is relatively long, an appreciable amount of vaporized fuel is present in the chamber at the time of ignition. As a result, combustion proceeds very rapidly, leading to high local flame temperatures and NO_x formation rates. Retarding the injection timing tends to reduce the NO_x emissions somewhat at the expense of a lower smoke and limited engine-power level (Ref. 6-23).

Another important advantage of the divided chamber diesel engine is its relatively low peak combustion pressure. Because of these reduced pressure levels, divided-chamber engines have some weight advantage over open-chamber designs.

6.3.3 Exhaust Emissions

Although diesel engines as a group are low polluters, they have a poor public image because of the odor and visible smoke emitted by some diesels under certain operating conditions. In addition to odor and smoke, diesel exhaust emissions include hydrocarbons (HC), carbon monoxide (CO), oxides of nitrogen (NO_x), oxides of sulfur (SO_x), and certain oxygenates including aldehydes and acrolein.

The HC emissions are primarily the result of incomplete combustion of fuel droplets in the cool regions adjacent to the cylinder walls and in zones of poor mixing and, to a lesser extent, due to combustion of lubricating oil. In general, the HC species found in diesel exhaust cover a wide range of molecular weights, including species that are heavier than those contained in the fuel. To a large extent, the level of HC emissions is related to the design and operational characteristics of the fuel injection system and combustion chamber. With good spray atomization and mixing, HC emissions are generally very low.

In the main, the CO emissions from diesel engines are very low because of the large quantity of excess air available for combustion under all operating conditions. Therefore, the level of CO emitted is related primarily to the degree of fuel-air mixing achieved in the cylinders.

NO_x represents the major pollutant specie emitted from diesel engines. The amount of NO_x formed in the combustion process is kinetically controlled and depends upon the peak combustion temperature, local oxygen and nitrogen concentrations and the residence time of the combustion products in the chamber. Also, fuel-bound nitrogen can have a significant effect on the total NO_x emissions.

Oxides of sulfur emissions are directly related to the sulfur content in the fuel. Control of SO_x is achieved by limiting the amount of sulfur in the fuel.

Normally, engine blowby represents a small contribution to the overall HC emitted from diesel engines. Control of blowby products in naturally aspirated diesels can be achieved by positive crankcase ventilation (Ref. 6-24).

Evaporative losses in diesels are very small because of the low volatility of diesel fuels. Also, since the injection system is a sealed unit, no fuel can escape through the injection nozzle following engine shutdown.

6.3.3.1 Heavy-duty Diesel Engine Emission Characteristics

Representative HC, CO, and NO_x specific mass emissions in terms of gram per brake horsepower-hour of heavy-duty, naturally aspirated and turbocharged open-chamber and prechamber diesels are presented in Table 6-2 for rated power and speed. These data represent average values and are based on tests conducted on a total of fourteen different engines.

The HC emissions listed for turbocharged open-chamber (direct injection) engines are higher than for naturally-aspirated engines of this type, although one would expect the opposite. The HC emissions of divided-chamber engines are especially low because of the high wall temperature of the prechamber and the high flow turbulence.

Table 6-2. Heavy-Duty Diesel Engine Emissions at Rated Power (Ref. 6-13)

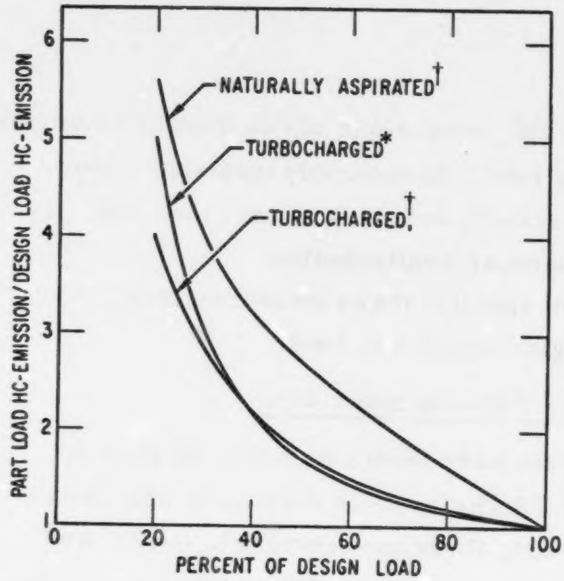
	Mass Emissions gm/bhp-hr)		
	HC	CO	NO_x
Open Chamber, naturally aspirated	1.1	7.2	7.5
Open Chamber, turbocharged	1.5	4.0	10.6
Divided Chamber, turbocharged	0.1	1.5	5.3
Divided Chamber, naturally aspirated*	0.1	3.3	4.0

*Mercedes Benz 220D light-duty diesel engine (Ref. 6-25)

The variations of CO with engine type are much smaller which is to be expected since CO is determined primarily by air-fuel ratio. Again, the prechamber engine type is the lowest emitter.

The NO_x emissions of turbocharged open-chamber engines are higher than for the other types and are comparable to spark ignition engines. This is primarily due to the high combustion temperatures achieved in turbocharged engines. Incorporation of an aftercooler tends to reduce the NO_x emissions for turbocharged engines.

The part-load HC, CO, and NO_x mass emissions of diesel engines operated at rated speed are presented in Figure 6-15 in terms of the ratio of the emissions at part load to the full-load emissions (Ref. 6-13). As illustrated, the HC emissions increase with decreasing engine load for all three engine types. The CO emissions of turbocharged divided-chamber engines increase steadily when engine load is reduced, while the naturally-aspirated and the turbocharged open-chamber diesels show an initial



a. Hydrocarbon emission, steady state part-load, rated speed.

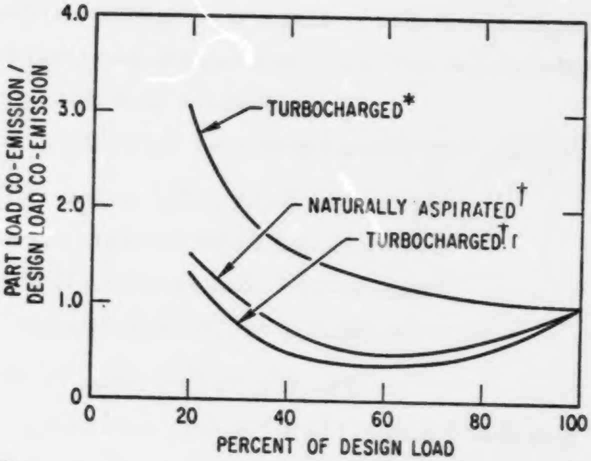
† OPEN CHAMBER

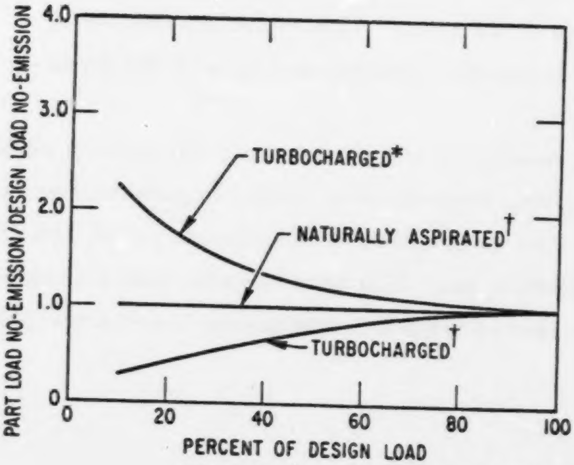
* DIVIDED CHAMBER

b. Carbon monoxide emission, steady state part-load, rated speed.

† OPEN CHAMBER

* DIVIDED CHAMBER





c. Nitric oxide emission, steady state part-load, rated speed

† OPEN CHAMBER

* DIVIDED CHAMBER

Figure 6-15. Four-Cycle Diesel Engines (Ref. 6-13)

reduction followed by an increase. The NO_x emissions of the divided-chamber diesel increase steadily with decreasing load. In naturally-aspirated open-chamber engines, the NO_x remains essentially constant at part load and decreases for the turbocharged open-chamber configuration.

In general, slightly lower specific mass emissions are obtained by simultaneously reducing engine speed and load.

6.3.3.2 Light Duty Diesel Vehicle Gaseous Emissions

A number of emission tests have been conducted on several diesel engine powered vehicles by the EPA (Refs. 6-26 through 6-28), Southwest Research Institute (SWRI, Ref. 6-29), Daimler-Benz (Ref. 6-12), and Peugeot (Ref. 6-30). To provide a direct comparison of the emissions from equivalent diesel and spark ignition automobiles, SWRI has also tested a Mercedes 220 gasoline engine powered car.

The emission test data presented next are based on the following four diesel-powered light-duty vehicles:

Mercedes Benz 220D sedan (EPA, SWRI, Daimler-Benz)
Opel Rekord 2100D sedan (EPA)
Peugeot 504 Diesel sedan (EPA, Peugeot)
Ford pickup truck with Nissan Diesel (EPA)

The Mercedes Benz 220D is a four-door sedan, powered by an overhead cam, 134-CID four-cylinder, four-stroke precombustion chamber diesel rated at 65 horsepower (SAE) and 4200 rpm. The vehicles tested by the EPA and SWRI were equipped with automatic transmission and the dynamometer weight was set at 3,500 pounds.

The Opel Rekord 2100D tested by the EPA was a four-door sedan, powered by a water-cooled, four-cylinder, four-stroke diesel engine utilizing Ricardo swirl chambers (Ref. 6-31). The engine has a displacement of 128 CID and develops 68 horsepower (SAE) at 4200 rpm. The test vehicle had a three-speed automatic transmission and was tested with a 3,000-pound inertia weight.

The Peugeot 504 diesel tested by the EPA was a four-door sedan with a water-cooled, four-cylinder, four-stroke, 128 CID swirl chamber engine rated at 65 horsepower (DIN) and 4,200 rpm, and a four-speed manual transmission. The tests were conducted with a 3,000-pound inertia weight.

The diesel-powered, light-duty truck tested by the EPA was a 1973 model year F-250 vehicle, retrofitted with a Chrysler/Nissan four-stroke, six-cylinder, 198-CID swirl-chamber diesel engine. The vehicle had a four-speed manual transmission and was tested with a 4,500-pound inertia weight.

The available HC, CO, and NO_x emission data for light-duty diesel-powered vehicles tested over the Federal Driving Cycle are presented in Table 6-3. The data represent average values from a series of runs conducted on the various vehicles. For comparison, the emissions from the gasoline engine powered Mercedes Benz 220 automobile, as tested by SWRI, are 2.7 gm/mi HC, 32.3 gm/mi CO, and 3.6 gm/mi NO_x.

As indicated, the Mercedes Benz 220D and the Opel Rekord 2100D meet the 1975 statutory emission standards. On the Mercedes 220D, the average HC and CO emissions are about 50 percent below the 1975 statutory standards. The average NO_x is approximately 50 percent below the 1975 statutory standard, but four times the statutory limit for 1976. The Daimler-Benz data are in reasonable agreement with the EPA and SWRI values. Modifications to the fuel injection system of this vehicle resulted in a 25 percent reduction of HC and CO, with no change in NO_x.

On the Opel, the CO and NO_x emissions are about 50 percent of the 1975 statutory level. However, the average HC is very close to this standard, with two tests indicating higher values and two indicating lower values. The average NO_x is about three times the 1976 statutory limit.

The average HC emissions from the Peugeot 504 diesel, as tested by the EPA, exceeded the 1975 standards. However, Peugeot states that the emissions could be substantially reduced by modifying the fuel injection equipment (Ref. 6-30). The average CO emissions exceed the 1975-76 statutory standard very slightly, while the NO_x emissions are about

Table 6-3. Average Light-Duty Diesel Vehicle Emissions (1975 Federal Test Procedure)

Test Vehicle	Test Laboratory	Test Procedure	Emissions, gm/mi			Reference
			HC	CO	NO_x	
Mercedes Benz 220D	EPA	1975 FTP	0.34	1.42	1.43	6-28
Mercedes Benz 220D (Modified Injection)	EPA	1975 FTP	0.28	1.08	1.48	6-28
Mercedes Benz 220D	SWRI	1972 FTP	0.87	1.62	1.83	6-29
Mercedes Benz 220D	Daimler-Benz	1975 FTP	0.15	2.3	1.45	6-12
Opel Rekord 2100D	EPA	1975 FTP	0.40	1.16	1.34	6-28
Peugeot 504 Diesel	EPA	1975 FTP	3.11	3.42	1.07	6-28
Peugeot 504 Diesel	Peugeot	1975 FTP	2.50	2.20	1.30	6-30
Ford/Nissan Diesel	EPA	1975 FTP	1.70	3.81	1.71	6-27

30 percent below the 1975 standard. The HC and CO data provided by Peugeot are substantially lower than the EPA data, while NO_x is slightly higher.

The modified Ford pickup truck with the Nissan diesel engine shows average NO_x emissions approximately 45 percent below the 1975 standard. The CO emissions are about 10 percent above the standard, while HC is about four times higher than the 1975-76 standard.

Emission data taken by Ricardo Engineers on a number of European diesel-powered automobiles tested in accordance with the 1975 Federal Driving Cycle are listed in Table 6-4. These data are in reasonable

Table 6-4. Diesel Automobile Emissions (Ref. 6-32)

Fuel Injection System	Emissions,* gm/mi		
	HC	CO	NO_x
Standard	0.17	1.6	1.2
Optimized Injection Timing	0.17	1.6	0.7
Optimized Injection Timing + EGR	0.17	2.5	0.4

*1975 Federal Test Procedure

agreement with the previously discussed data. Also presented in the table are projected emissions to be achieved by optimized timing and EGR, at some increase in fuel consumption. Ricardo estimates that achievement of the 0.7 gm/mi NO_x level would result in a 10 to 12 percent loss in fuel economy.

When evaluating these emissions, one must consider the small data sample available for each vehicle type. Therefore, an assessment of car-to-car variability and the effects of mileage accumulation on emissions is not possible. Also, it should be noted that the engine power of some of the vehicles was insufficient to negotiate the first and second acceleration modes of the Federal Driving Cycle.

6.3.3.3 Smoke and Particulate Emissions

The smoke emitted by diesel-powered vehicles can be grouped into three categories -- white, blue, and black. White smoke or cold smoke, which usually appears in the exhaust during an engine cold start, consists of partially oxidized fuel droplets combined with other compounds, such as aldehydes. It is formed in the wall region of the cylinders where the temperatures are not high enough to ignite the fuel. Blue smoke is emitted by some diesels as a result of the combustion of excessive amounts of lubricating oil. Black smoke, or hot smoke, consists of agglomerated carbon particles and partly burned hydrocarbon species formed as a result of incomplete combustion in fuel-rich regions of the combustion chamber (Ref. 6-2).

Smoke data from four naturally aspirated, four-stroke diesel engines are depicted in Figure 6-16. The sharp increase of the smoke level of engines A and B at high loads may be caused by poor injector performance, resulting in inadequate fuel atomization and fuel impingement on the cylinder walls. In general, turbocharged engines exhibit lower smoke levels at full load because of the higher excess air available (Ref. 6-25). Springer and Ashby conducted smoke tests on a diesel-powered Mercedes Benz 220D automobile in accordance with the Federal Smoke Test Procedure for heavy-duty vehicles (Ref. 6-33). The results from these tests are presented in Table 6-5, showing smoke opacity during low gear acceleration ("a"-factor), lug-down ("b"-factor), and at peak smoke conditions ("c"-factor). As indicated, the engine meets the 1974 federal heavy-duty smoke standards.

Particulate emissions from the Mercedes 220D engine determined by the Dow Chemical Procedure are listed in Table 6-6. The average particulate emission of 0.73 g/mi is about 2 to 3 times higher than for a gasoline engine using leaded fuel and about 10 to 20 times higher than for nonleaded gasoline. The material collected was pitch black and very fine (Ref. 6-25). Tests conducted by Daimler-Benz indicate average particulate emissions of 0.45 g/mi (Ref. 6-12).

Smoke and particulate emissions from diesel engines can be minimized by several means, including proper injector design and maintenance, engine derating, smoke suppressant additives in the fuel, and possibly filters and particulate traps (Ref. 6-34). The additives can be divided into two groups. One group is represented by the detergent types which maintain a clean fuel injection system. The other group consists of metal-containing materials which reduce the ignition temperature of the carbon and promote its oxidation.

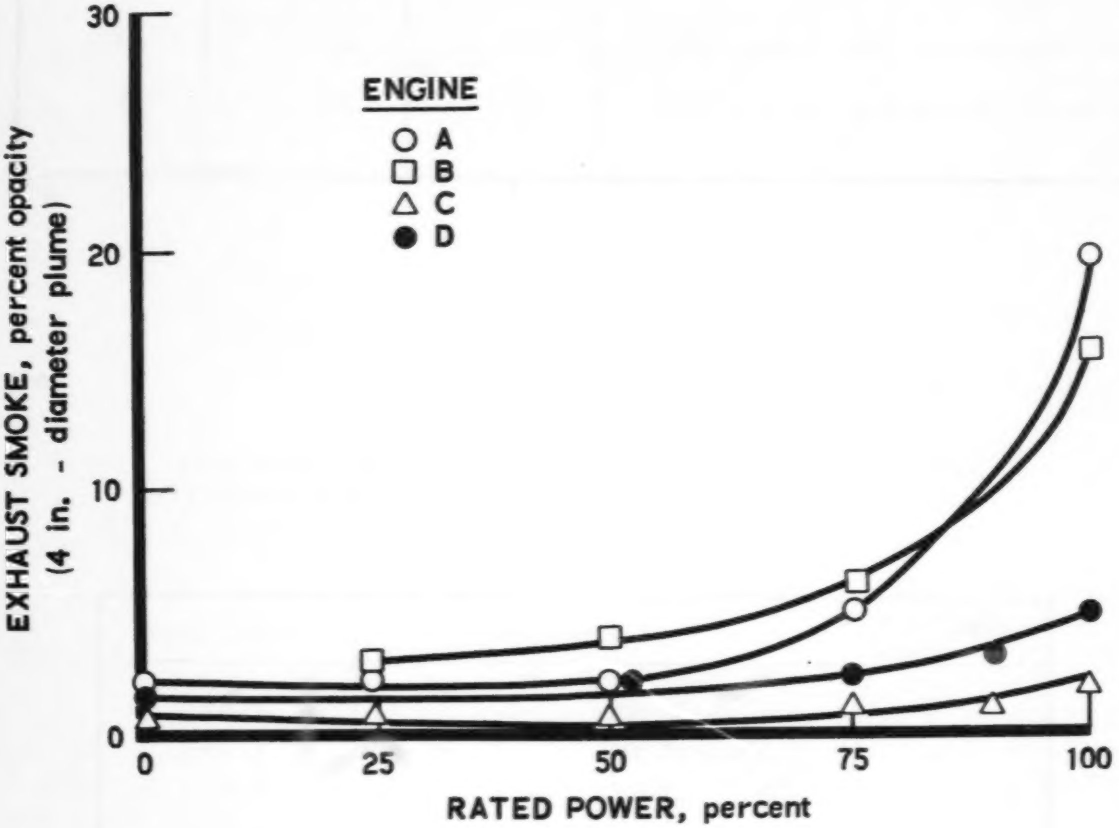


Figure 6-16. Exhaust Smoke vs Output Power for Four Open-Chamber, Naturally Aspirated, Heavy-Duty Diesel Engines (Ref. 6-25)

Table 6-5. Smoke Emissions from Mercedes Benz 220D Automobile, Simulated Federal Smoke Tests (Ref. 6-33)

	Percent Opacity		
	"a" factor	"b" factor	"c" factor
Average from 4 runs	10.8	9.4	15.1
Federal Heavy Duty Limit, 1970	20	40	-
Federal Heavy Duty Limit, 1974	15	20	50

Table 6-6. Particulate Emissions from Mercedes Benz 220D Automobile, Dow Chemical Procedure, gm/mile (Ref. 6-33)

Sample Flow, cfm	Filter Type	Mass, gpm
4	Fiberglass	0.66
1	Fiberglass	0.80
1	Fiberglass	0.75
1	Anderson + Millipore	0.73
	Average	0.73

Some work with fuel additives containing barium has been conducted by several investigators (Refs. 6-35 and 6-36). In all cases, black diesel smoke was substantially reduced with relatively small amounts of additives. However, there may be health effects resulting from such additives, and they are currently not recommended for use.

6.3.3.4 Emission Control Techniques

A number of emission control techniques and devices have been considered by many investigators for potential use in diesel engines. These include injection timing retardation, valve overlap, EGR, water injection, catalysts, thermal reactors, and turbocharging. These approaches are briefly described in the following paragraphs.

Injection timing retardation in general has a beneficial effect on NO_x emissions at the expense of higher HC and CO and reduced fuel economy. However, since HC and CO emissions are inherently low in diesels, a tradeoff between these species might be feasible (Refs. 6-23, 6-37, and 6-38).

Modification of the valve overlap can reduce the NO_x emissions by reducing the degree of engine scavenging. However, the benefits derived from this approach are apparently rather limited (Refs. 6-12 and 6-24).

Exhaust gas recirculation (EGR) appears to be the most promising technique for diesel NO_x control. As in spark ignition engines, EGR acts as a charge diluent and reduces the peak temperature in the combustion chamber. Normally, this results in some increase in fuel consumption and HC, CO, and smoke emissions (Refs. 6-2, 6-23, and 6-37). Conversely, test data from a Mercedes Benz 220D vehicle presented in Figure 6-17 indicate no change in HC, CO, and smoke for EGR rates up to about 25 percent (Ref. 6-12). Utilization of 25 percent EGR has resulted in a 25 to 50 percent reduction in NO_x, depending upon the operating conditions of the engine. Similar results have been reported by Marshall and Fleming (Ref. 6-37). According to Daimler-Benz, there is apparently no loss in part load fuel economy with EGR (Ref. 6-12). However, deposit buildup and corrosion in the intake ports

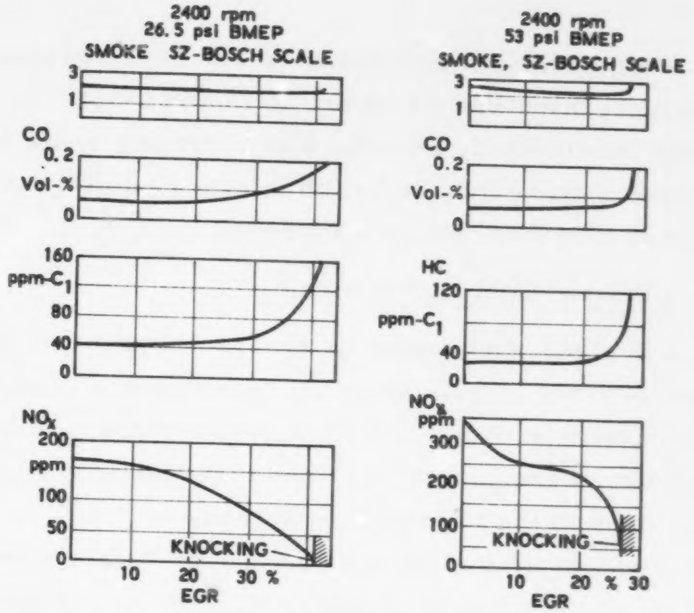


Figure 6-17. Effect of Exhaust Gas Recirculation on the Emissions from a Light-Duty Diesel Engine (Ref. 6-12)

have been experienced by Daimler-Benz which might create a valve durability problem. Caterpillar has encountered similar problems (Ref. 6-24).

As with EGR, water injection into the engine intake is an effective means to reduce NO_x emissions from diesel engines (Refs. 6-2, 6-29, and 6-36). This is illustrated in Figure 6-18 showing NO_x emissions, from a turbocharged heavy-duty diesel engine operated at 2,200 rpm, as a function of water-injection rate. Since the inducted water vaporizes and displaces air, the air flow rate into the engine decreases, and this may result in an increase in smoke. Potential problem areas related to water injection include corrosion of the induction system and the intake valves.

The HC and CO emissions from diesels can be further reduced by means of catalytic converters installed in the engine exhaust system (Ref. 6-2). However, in view of the already low HC and CO emissions of diesel engines, this approach is probably not very cost-effective. Control of

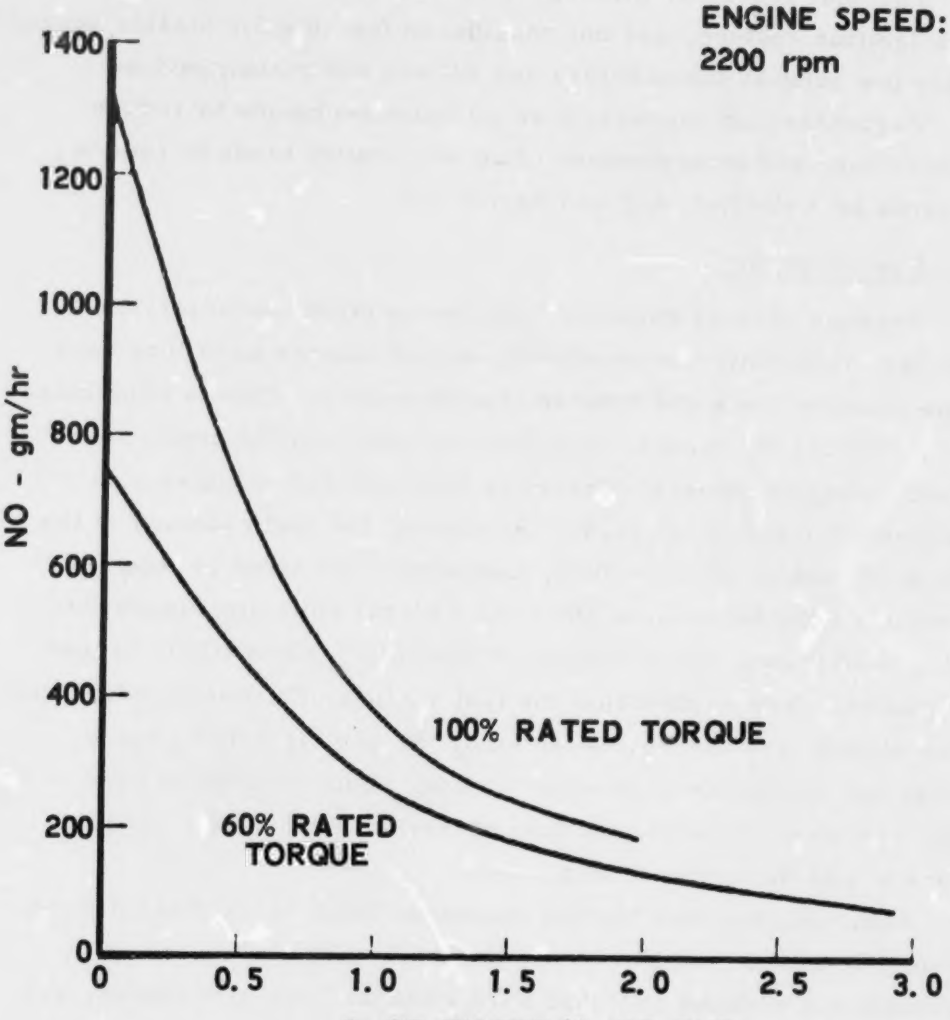


Figure 6-18 . Effect of Water Induction Rate on the NO Emissions of a Turbocharged Diesel Engine (Ref. 6-24)

NO_x by means of reducing catalysts is not possible for diesels because of the excess oxygen present in the exhaust.

Thermal reactors, although quite effective in reducing HC and CO from spark ignition engines, are not considered feasible for diesels because of the relatively low exhaust temperature and HC and CO concentrations.

Turbocharging represents an effective technique to reduce HC and CO emissions, and incorporation of an aftercooler tends to reduce the NO_x emissions as well (Ref. 6-2 and Ref. 6-23).

6.3.4 Fuel Economy

Because of their superior fuel consumption characteristics, combined with high reliability and durability, diesel engines have long been favored for use in many truck and industrial applications. This is illustrated in Figure 6-19, showing the steady-state fuel economy of 1972 model year diesel and gasoline engine powered Mercedes 220D and 220 vehicles as a function of level-road speed (Ref. 6-39). As shown, the fuel economy of the diesel vehicle at 30 mph is about 50 mpg, compared with about 25 mpg for the gasoline vehicle adjusted to meet the 1972 Federal emission standards. With increasing road speed, the advantage of the diesel with respect to fuel economy diminishes. It is evident that the fuel savings of a diesel, compared with a gasoline engine, are strongly affected by the vehicle driving cycle. The fuel savings are negligible at freeway speeds, about 30 percent over country roads, and about 50 percent in taxi operation (Ref. 6-32). At idle, the fuel savings are of the order of 70 percent.

Fuel economy data for the Mercedes Benz 220D, Opel Rekord 2100D, Peugeot 504 diesel, and Ford/Nissan diesel vehicles, as tested by the EPA in accordance with the 1975 and 1972 Federal Test Procedures, are presented in Table 6-7 (Refs. 6-26 through 6-28). The figures were computed by the EPA from the measured HC, CO, and CO_2 exhaust concentrations using the carbon balance method. Also listed in the table are the fuel economy values for the 1973 model year gasoline engine powered Mercedes Benz 220, the Peugeot 504, and the average 3,000-pound, 3,500-pound and 4500-pound 1973 automobiles as determined by the EPA from certification

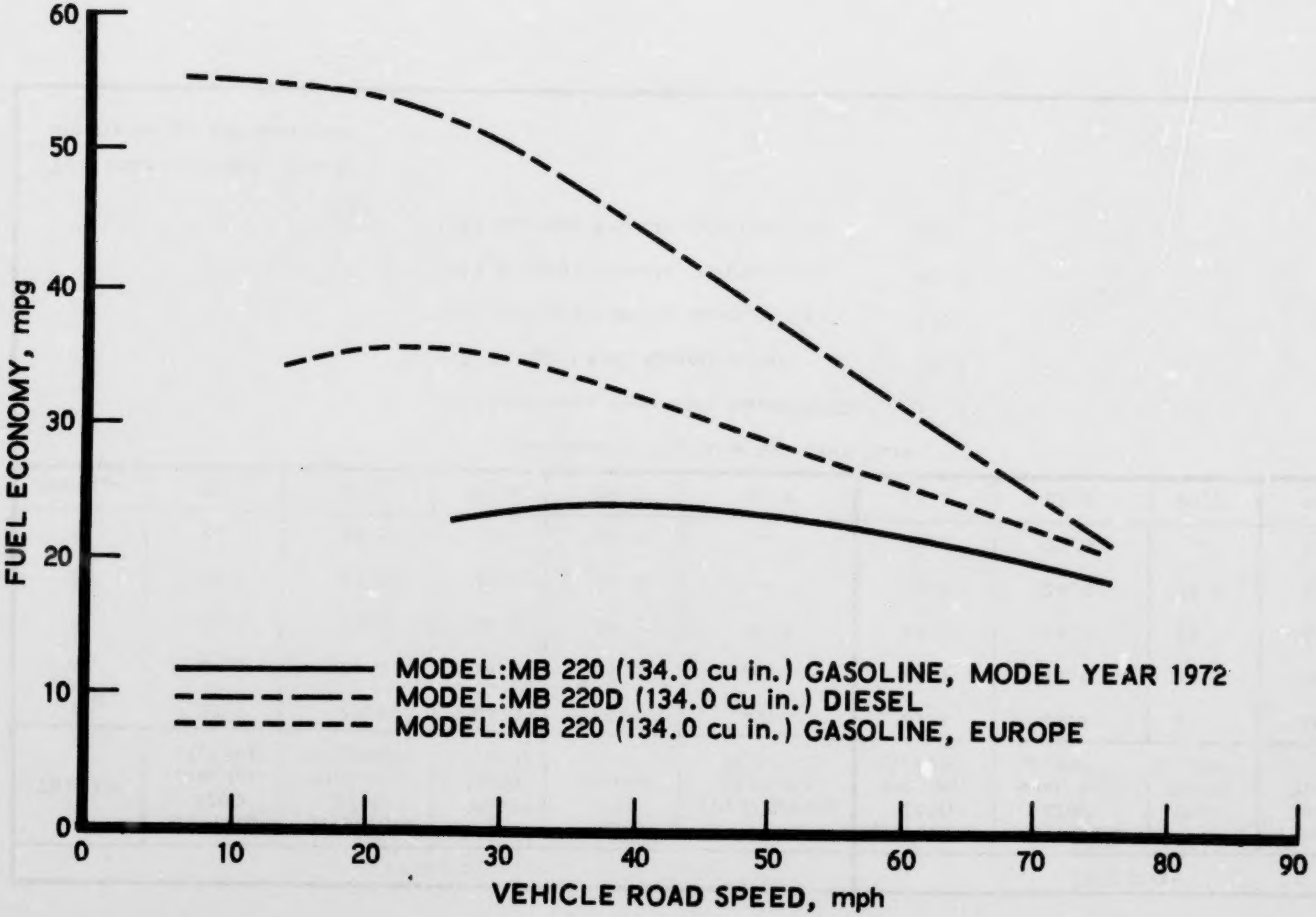


Figure 6-19. Fuel Economy versus Vehicle Speed for Diesel and Gasoline Automobiles (Ref. 6-39)

Table 6-7. Light-Duty Diesel and Gasoline Vehicle Fuel Economy, miles/gallon (Refs. 6-26, 6-27, 6-28)

	1975 FTP					1972 FTP			
Test No.	Mercedes 220D Std. Inj. (3,500)*	Mercedes 220D Mod. Inj. (3,500)*	Opel 2100D (3,000)*	Peugeot 504 Diesel ( )**	Ford/Nissan Pick-Up (4,500)*	Mercedes 220D Std. Inj. (3,500)*	Mercedes 220D Mod. Inj. (3,500)*	Opel 2100D (3,000)*	Peugeot 504 Diesel ( )**
1	23.7	23.8	23.9	25.4	22.2	23.3	22.8	23.3	24.3
2	22.6	24.4	24.3	25.2	21.3	22.3	23.8	23.5	24.3
3	23.8	25.3	23.2	24.1	20.7	33.3	24.0	22.7	23.3
4	24.0	24.9	23.7	25.2	-	23.9	24.0	23.1	24.2
5	24.0	24.5	-	26.2	-	23.5	23.6	-	25.1
Average	23.6	24.6	23.8	25.2	21.4	23.3	23.6	23.2	24.2

Comparative Gasoline Powered Cars

1973 Mercedes Benz 220, Gasoline Car	13.2
1973 Peugeot 504, Gasoline Car	17.0
1973 Average 3000-lb Gasoline Car	15.6
1973 Average 3500-lb Gasoline Car	13.9
1973 Average 4500-lb Gasoline Car	10.1

*Test inertia weight, pounds

**Inertia weight not available

data (Ref. 6-40). As indicated, the fuel economy of the diesel-powered vehicles is substantially higher than that of equivalent gasoline automobiles. For instance, the Mercedes Benz 220D diesel with standard fuel injection has a fuel economy of 23.6 mpg, compared with 13.2 mpg for the gasoline engine powered Mercedes 220 with automatic transmission and 13.9 mpg for the average 3,500-pound 1973 model year certification vehicle. These differences correspond to fuel savings of about 41 percent and 44 percent, respectively. The fuel economy values for the Opel 2100D and Peugeot 504 diesel automobiles are 23.8 mpg and 25.2 mpg, respectively, compared with 15.6 mpg for the average 3000-pound 1973 model year car. The corresponding fuel savings are about 42 percent and 38 percent, respectively. For the Ford/Nissan diesel powered pickup truck, the measured average fuel economy is 21.4 mpg, which is more than twice that of the average 4,500-pound 1973 model year automobile.

However, when comparing the fuel economy of current diesel and gasoline powered automobiles, an adjustment must be made to account for the higher power and torque output of the gasoline engine. This is illustrated in Figure 6-20, showing the fuel economy of 3,000-pound and 3,500-pound 1973 model year certification vehicles as a function of rated engine power. Since the fuel economy increases with decreasing power, the fuel savings realized with the diesel over gasoline engines of comparable performance would be somewhat lower than noted above.

6.3.5 Noise

In general, diesel engines are inherently noisier than gasoline engines, in particular the open-chamber designs most frequently used in truck and industrial applications. The noise is especially strong at idle and during a cold start of the engine. Although the noise levels of light-duty automotive diesel vehicles are generally considered acceptable in the normal operating regime of the engine, the noise from these engines is noticeably higher than that of modern V-8 gasoline engines (Refs. 6-12, 6-41, and 6-42).

The principal contributors to diesel noise are the combustion pressure transients, piston slap, and timing gear impacts. Other sources include valve gear and fuel injection system impacts (Ref. 6-42).

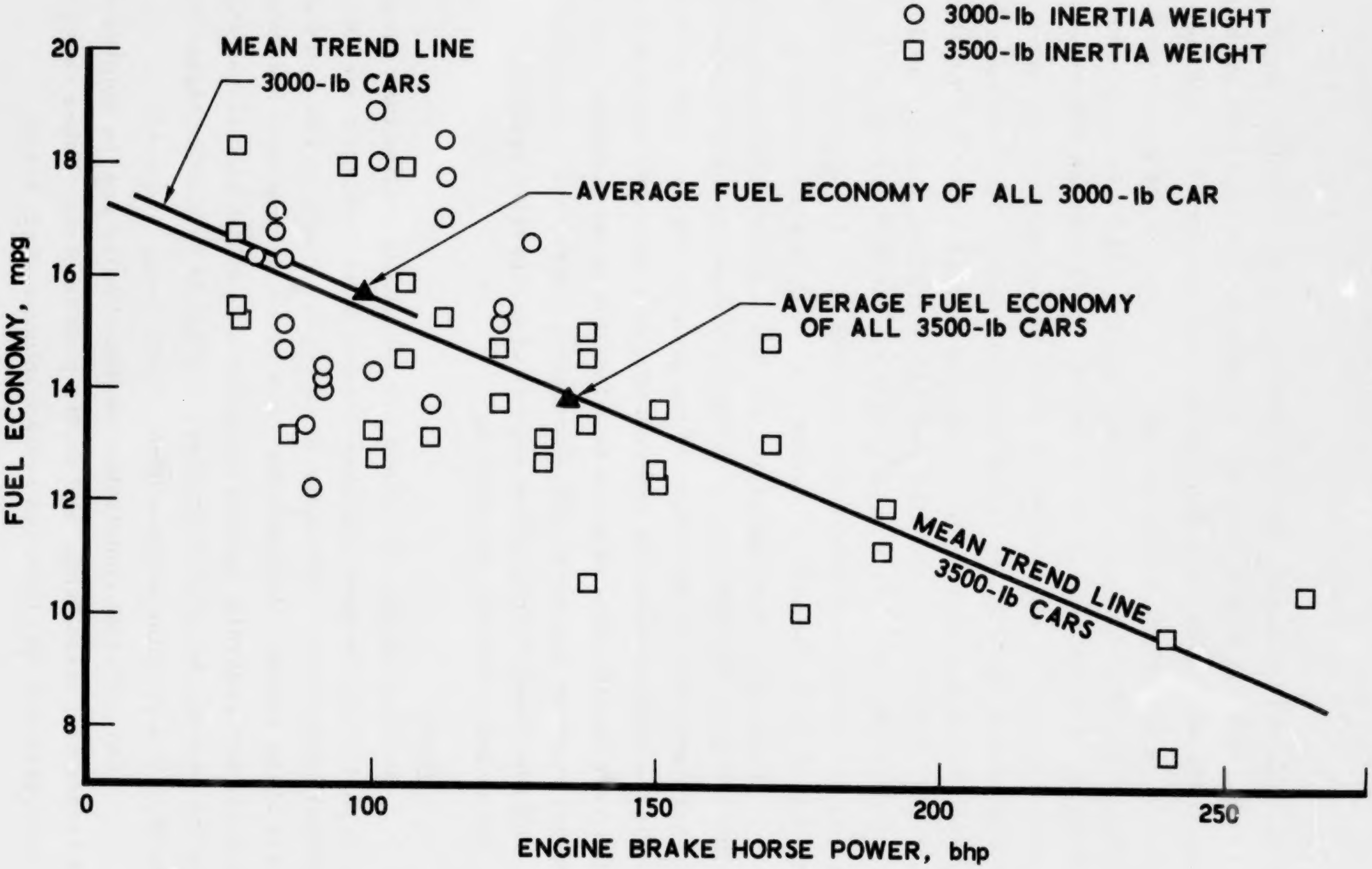


Figure 6-20. Fuel Economy versus Engine Rated Power, 1973 Model Year Certification Vehicles

Exterior and interior noise levels reported by Daimler-Benz for its Mercedes 220D diesel and Mercedes 220 gasoline vehicles are shown in Table 6-8 and Figure 6-21. With the exception of a few operating points, the diesel noise is of the order of 2 to 3 dB(A) higher than the noise of the gasoline engine, as shown.

Interior noise data from Peugeot 504 diesel and gasoline engine powered automobiles are listed in Table 6-9 for a number of steady-state operating conditions of the vehicles. Again, the diesel engine tends to be slightly noisier than the gasoline engine (Ref. 6-30).

Several approaches are currently being considered by industry to reduce the noise of diesel engines. These include reduction of component clearances and structural changes on the engine and engine compartment to modify their vibration characteristics (Refs. 6-12 and 6-42). Intake air heating might prove effective to reduce the noise at idle.

6.3.6 Odor

Diesel odor has long been recognized as a very undesirable exhaust emission product. Although a considerable amount of research has

Table 6-8. Exterior Vehicle Noise, Mercedes Benz 220D Diesel and 220 Gasoline Automobiles (Ref. 6-12)

	Exterior Noise, dB(A)	
	Mercedes 220D	Mercedes 220
Engine Start	65	60
Idle	56	51
Normal Vehicle Start	72	69
Fast Vehicle Start	78	80
30 mph (third gear)	75	70
44 mph (fourth gear)	83	81
62 mph (fourth gear)	79	75

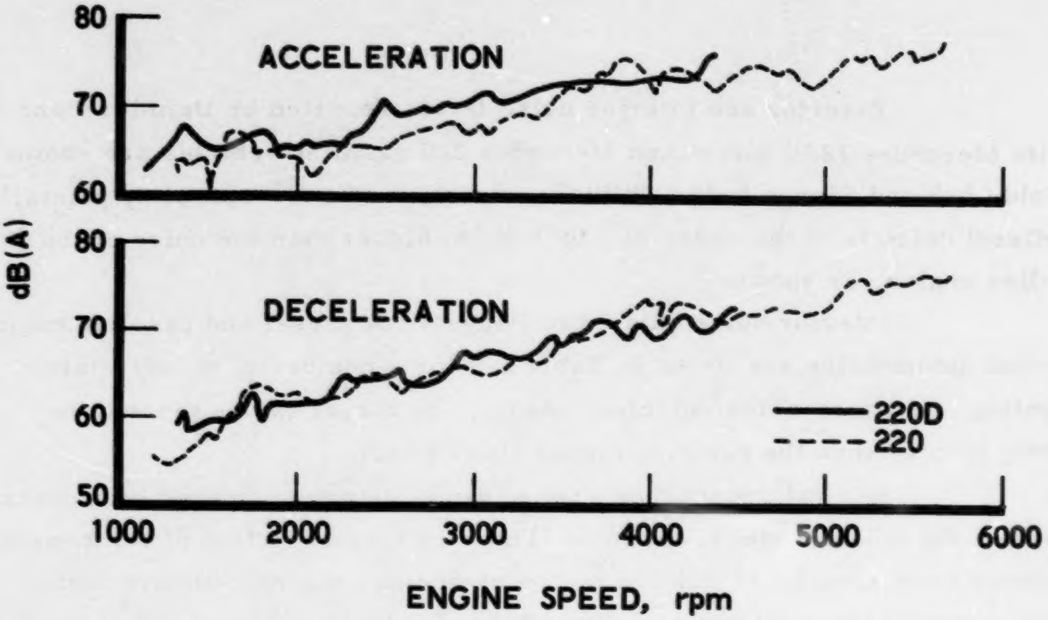


Figure 6-21. Interior Vehicle Noise, Mercedes Benz 220D Diesel and 220 Gasoline Automobiles (Ref. 6-12)

Table 6-9. Interior Noise, Peugeot 504 Diesel and Gasoline Automobiles (Ref. 6-30)

Speed	Peugeot 504 * Gasoline Sedan dB(A)	Peugeot 504 * Diesel Sedan dB(A)
Idle	50	54
31 mph	65	67
50 mph	70	70
62 mph	72	72
74 mph	76	78
87 mph	79	-
Maximum Speed	82	-

*Noise measurements taken in the interior of the vehicle at constant speed in fourth gear in dB(A) been under way for some time, progress toward determination of the cause of the odor has been very slow because of the complexity of the heterogeneous combustion occurring in diesels, combined with the lack of accurate instrumentation (Refs. 6-43 and 6-44).

The diesel odor is known to be related to the combustion process, chamber shape, and (to a lesser degree) fuel type and fuel composition.

Diesel exhaust contains small quantities of a large number of unburned and partially burned hydrocarbon compounds. Some of these compounds have strong odors, even in very low concentrations, and mixtures of these materials are believed to be responsible for the formation of the typical diesel odor (Ref. 6-2). Two major types of odor have been identified, one being an "oily-kerosene" type apparently caused by aromatic hydrocarbons and the other being of the "smokey-burnt" type, and probably caused by partially oxidized compounds. Barnes (Ref. 6-45) has concluded that diesel odor is formed by partial oxidation reactions in the fuel-lean regions of the engine.

In the absence of reliable instrumentation, odor has been evaluated in the past by specially selected human panels trained to recognize both odor quality and intensity. This panel classifies the odor by comparing it with 28 different odor qualities and intensities supplied from the bottles of the Quality/Intensity Evaluation kit. The overall diesel composite rating "D" ranges from D-1 to D-12, with D-12 representing the strongest odor. Burnt-smokey quality "B," aromatic quality "A," oily quality "O," and pungent quality "P" have a range between 1 and 4. More recently, a diesel odor analysis instrument has been developed by A. D. Little, Incorporated, which is designed to permit direct measurement of diesel odor levels (Ref. 6-46). Evaluation of the instrument is in progress.

The effect of engine operating condition on odor intensity can vary markedly for different engines. For naturally aspirated engines run at rated speed, the minimum odor intensity seems to occur in the mid-power range of the engine; in turbocharged engines, odor intensity remains essentially constant over the power range. This trend is believed to be the result of a reduction in the ignition delay time occurring with rising power output due to an increase in the intake and compression temperatures. Since odorants are thought to be formed by partial oxidation of the fuel prior to ignition, the constant odor intensity pattern would follow from this effect (Ref. 6-35).

Odor data reported by Springer for a Mercedes Benz 220D diesel vehicle are listed in Table 6-10 (Ref. 6-29). The data show no clear trend of odor intensity with engine speed and load. The highest odor intensity was recorded during engine deceleration. In general, the odor levels of the Mercedes engine are quite comparable to the levels observed on heavy-duty diesel engines, although the character of the odor appears somewhat different. Other exhaust emission species, including HC, CO, NO_x, aldehydes, formaldehyde, and acrolein, measured along with the odor, are listed in Table 6-10. As indicated in the table, odor intensity in the diesel exhaust correlates reasonably well with HC and acrolein emission concentrations. It should be noted, however, that the mass emissions of the aldehydes, formaldehyde, and acrolein were much lower for the Mercedes Benz 220D diesel than for the 220 gasoline engine. These results are difficult to explain and require further study.

Several odor-abatement approaches have been considered by industry, including injector modifications, catalytic mufflers, and fuel additives. Utilization of a needle-valve injector with a very small dribble volume in place of a check-valve-type injector has resulted in a substantial reduction of the odor intensity from two General Motors two-cycle, GV-71 diesels. Also, catalytic mufflers have shown some benefit, but in early tests their effectiveness deteriorated rapidly with mileage accumulation. However, in view of the catalyst improvements achieved in the past few years, further work in the area of catalytic odor abatement might be desirable. With respect to fuel additives, some reduction in odor intensity has been demonstrated by Southwest Research Institute (Ref. 6-47). However, the benefits achieved to date are small and more research would be required to permit a meaningful feasibility assessment of this technique.

Table 6-10. Average Odor and Gaseous Emissions from Mercedes Benz 220D Automobile (Ref. 6-29)

Engine Speed RPM	Load Percent	Odor					Gaseous Emissions, ppm					
		Composite "D"	Burnt "B"	Oily "O"	Atomatic "A"	Pungent "P"	HC	CO	NO_x	Acrolein	Formal-dehyde	Aliphati Aldehy
2500	0	6.0	2.0	1.4	1.1	1.5	980	7.3	78	7.7	15.4	28.2
2500	50	3.8	1.1	1.0	0.8	0.7	71	231	339	1.4	15.2	21.1
2500	100	4.0	1.2	1.0	0.9	0.8	64	209	505	1.0	13.7	14.1
4200	0	4.4	1.3	1.1	1.0	1.0	89	348	224	1.2	7.9	12.6
4200	50	6.1	2.1	1.3	1.1	1.6	739	1155	528	6.3	20.0	35.1
4200	100	5.5	1.9	1.3	1.1	1.3	60	968	598	1.5	21.7	25.1
Idle	0	3.9	1.1	1.0	1.0	0.8	105	168	124	1.6	12.0	19.0
Acceleration	-	5.2	1.8	1.2	1.0	1.2	-	-	-	-	-	-
Deceleration	-	6.8	2.1	1.7	1.2	1.8	-	-	-	-	-	-

6.3.7 Reliability and Durability

Because of a lack of applicable test data, a quantitative evaluation of the reliability and durability characteristics of light-duty automotive diesel engines is not possible. However, in general terms, the reliability of diesels is quite high and probably superior to equivalent gasoline engines (Ref. 6-12). The excellent reliability of heavy-duty automotive and industrial diesels is well known.

Although engine durability depends on many parameters (such as individual driving patterns; maintenance programs; and, to a lesser extent, fuel quality) there are indications that light-duty automotive diesels are more durable than equivalent gasoline engines. In truck and industrial applications, the ruggedness and superior durability characteristics of the diesel engine have been adequately demonstrated. Many truck diesels have been operated over 300,000 miles and more with minimum maintenance.

6.3.8 Maintenance

With respect to engine maintenance, the diesel engine has definite advantages over the gasoline engine simply because it has fewer parts that require periodic adjustment. For instance, the diesel has no carburetor and ignition system; hence tuneups and replacement of spark plugs, points, and condenser are not needed. This results in substantial cost savings.

The scheduled maintenance on diesels includes periodic valve adjustments, oil and oil filter changes, and inspections and adjustments of the fuel injection system. According to Daimler-Benz, the injection system in its 220D automobile is practically maintenance-free (Ref. 6-12). Generally, maladjustment of the fuel injection system or carbon buildup on the nozzles is accompanied by the appearance of visible smoke in the exhaust (along with an increase in HC and CO emissions) and, in severe cases, by a noticeable loss in power. As a result, it is more likely that the owner of a diesel vehicle would take the necessary steps to eliminate the problem than the owner of a gasoline car where quite frequently, fouling of certain components might greatly increase the emissions without affecting engine performance.

6.3.9 Safety

Since diesel fuel is much less volatile than gasoline, formation of explosive fuel air mixtures in enclosed areas due to fuel evaporation is essentially impossible. Also, in the event of an accident, the chance of an engine or grass fire resulting from spilled fuel is extremely remote. Therefore, diesel is considered to be a safer fuel than gasoline.

6.3.10 Driveability

A number of performance parameters have been identified for gasoline engine powered automobiles having an adverse effect on vehicle driveability. These include (1) engine stalling during start, (2) engine stumble, hesitation, surging, knocking, and backfire during acceleration, and (3) backfire and stalling during vehicle deceleration. In diesel engines, deficiencies of this type are not common; hence, driveability of diesel-powered automobiles should be excellent.

With respect to acceleration, available test data indicate that current diesel-powered automobiles accelerate much more slowly than gasoline vehicles of similar weight and engine size. For example, the Mercedes Benz type 220D automobile requires 41.5 seconds to accelerate from 30 mph to 70 mph, compared with 18.3 seconds for the Mercedes Benz 230 gasoline car adjusted to meet the 1975 Federal emission standards, and 12 seconds specified for the U.S. Department of Transportation's Experimental Safety Vehicle (Ref. 6-12). However, the acceleration capability of diesel vehicles could be substantially improved by supercharging.

6.3.11 Cost

The diesel-powered automobiles currently marketed in the United States by Daimler-Benz and in Europe by Daimler-Benz, Peugeot, and Opel are more expensive than their spark ignition counterparts. According to Daimler-Benz (Ref. 6-12), the cost differential is due to a number of unique additional components required in diesel engines, including the fuel injection system,pump drive, cylinder liners, special pistons, and glow plugs. In addition, the sturdier construction, the complex prechamber, and the larger starter motor and battery required in the diesel further contribute to the higher cost of this engine.

To illustrate this point, the purchase price of the diesel-powered Mercedes Benz 220D is currently $211 higher than the otherwise identical Mercedes 220 gasoline car. However, the higher initial cost of the diesel is counteracted by the lower fuel consumption and maintenance cost of this engine. Furthermore, the cost differential diminishes as more emission control equipment is added to the gasoline vehicle to meet future emission control standards.

According to Peugeot and Opel, the cost differential in Europe between diesel and gasoline engine powered automobiles is approximately $600 to $800. In California, the base price of the Peugeot 504 diesel sedan is $5800 as compared to $4830 for the standard 504 vehicle with a spark-ignition engine.

In the case of Daimler-Benz, the relatively small cost differential between the 220 series diesel and gasoline vehicles is due to the fact that the two engines were designed with commonality of parts as a prime consideration. Conversely, diesels manufactured by Peugeot and Opel are special designs, and their higher cost reflects the low-production volume of these engines.

Currently, no reliable data are available regarding the effects of turbocharging on the cost of light-duty diesel engines. In truck engines, the cost of the turbocharger is of the order of 15 percent of the cost of the engine. Based on a preliminary design study, Ricardo feels that a new light-duty diesel engine could be designed which would be acceptable to the U.S. consumer when installed in a medium-size automobile. The cost of this 100 to 130 hp engine on a mass production basis would be about 40 percent higher than that of an equivalent gasoline engine, resulting in a vehicle cost increase of about 5 percent (Ref. 6-32).

6.4 CURRENT STATUS OF TECHNOLOGY

Diesel engines are widely used in Europe and Japan for taxicabs and other light-duty vehicles, primarily because of their lower fuel consumption. However, the power output of these engines is considerably lower than that required for the heavier American vehicles, and the operating conditions are generally different.

The Mercedes Benz 220D diesel engine powered automobile has been marketed in the United States since 1956. Early in 1974, Peugeot of France offered its model 504 diesel passenger car for sale in this country. Conversely, General Motors (Opel of Germany) has no plans to enter the United States market with its diesel-powered automobile model.

Daimler-Benz of Germany has been producing diesel automobiles since 1936. To date, over one million of these vehicles have been sold to customers in over 100 countries. According to Daimler-Benz (Ref. 6-12), the diesel vehicles are marketed primarily in those areas of the world where fuel cost is high and where engine power is not a prime consideration. Between 1956 and 1972, a total of 53,331 diesel automobiles have been sold by Daimler-Benz in the United States, with more than 6,000 sold in 1972 alone. In 1971, Daimler-Benz manufactured over 99,000 Mercedes Benz 220Ds out of a total car production of about 284,000.

Peugeot introduced its diesel-powered automobile in 1959 and has since sold a total of about 460,000 vehicles. In 1972, diesel sales amounted to nearly 80,000 out of a total production of 671,000 vehicles.

The available test data indicate that the emissions of diesel-powered light-duty vehicles manufactured by Daimler-Benz and Opel are below the statutory 1975 Federal emission standards. NO_x levels of the order of 0.8 g/mi or even lower appear achievable by means of EGR or water injection, combined with injection system modifications.

The fuel economy (mpg) of the current light-duty diesel vehicles over the Federal Driving Cycle is between 50 percent and 70 percent better than that of the average 1973 certification vehicles tested at the same inertia weight.

6.5 PROJECTED STATUS

In the absence of adequate technical information on advanced light-duty, lightweight diesel engines, it is extremely difficult to make meaningful projections regarding the future use of advanced diesels in passenger cars.

According to Daimler-Benz (Ref. 6-12), the diesel engine should not be considered as a replacement of the gasoline engine, but rather a supplementary concept for specific applications where fuel economy and engine durability are of prime concern. However, with the advent of more stringent emission control requirements, and with the energy shortage in mind, interest in the diesel as an automobile power plant is expected to increase in the future. In spite of these developments, it will be very difficult to substantially increase the production volume of diesels within a reasonable time span. In this regard, Daimler-Benz feels that a minimum of 24 months would be required to increase its current production volume on the 220D vehicle. Conversely, a lead time of 5 to 6 years is projected by Daimler-Benz for the design and development of a new, more powerful diesel engine and for the construction of a production facility (Ref. 6-12).

It is difficult to ascertain whether the general public in the United States would ever be willing to accept the low power capability of the current diesel-powered automobile. Therefore, it appears appropriate to briefly dwell upon a number of approaches which could be pursued to increase the power output of diesel engines and to reduce their weight and bulk.

Turbocharging or supercharging is considered to be a feasible technique to increase the specific power output of diesels with a concomitant improvement in fuel economy. However, the current light-duty diesel engine designs may not be rigid enough to withstand the higher combustion pressures and temperatures encountered by adding a turbocharger. The stress problems in the cylinders, pistons, and crankshaft could be alleviated by reducing the compression ratio of the engine. Since the compression temperature decreases with decreasing compression ratio, engine starting becomes more difficult, especially at low ambient temperatures, and starting aids would have to be utilized such as glow plugs, intake air heaters, ether injection, preinjection of fuel, or even variable compression ratio pistons. This is illustrated in Figure 6-22, showing the compression temperature and the ignition temperature of diesel fuel and gasoline as a function of the compression ratio. Using diesel fuel and operating at normal ambient conditions, the diesel can maintain idle speed for compression ratios above about 12:1 (Ref. 6-15).

In general, turbochargers are not very effective at low engine speeds, and they have inadequate response to the load variations encountered in light-duty vehicle operation. These deficiencies are overcome by the Comprex supercharger, which has a much faster response and higher BMEP capability in the low speed regime of the engine. As a result, the low speed torque of the engine increases substantially, and this permits a reduction of the design peak power at a concomitant savings in engine weight and bulk. Additional weight savings might be possible by reducing the design life of the diesel engine, which is several hundred thousand miles in current designs.

Ricardo and Company Engineers of England, under contract to the EPA, is currently involved in a survey of the problem areas and the impact of diesels as power plants for passenger car applications. The parameters being considered by Ricardo include two stroke vs four stroke, naturally aspirated and turbocharged diesels, rotary diesels, EGR, startability, emissions, odor, and noise.

With regard to larger cylinder and engine displacement, it appears that this approach alone is not sufficient to achieve an adequate increase in engine power. Apparently, the requirement for good combustion and smoothness of operation limits the maximum cylinder volume of high-speed prechamber diesel engines to about 36 CID. Hence, four-cylinder

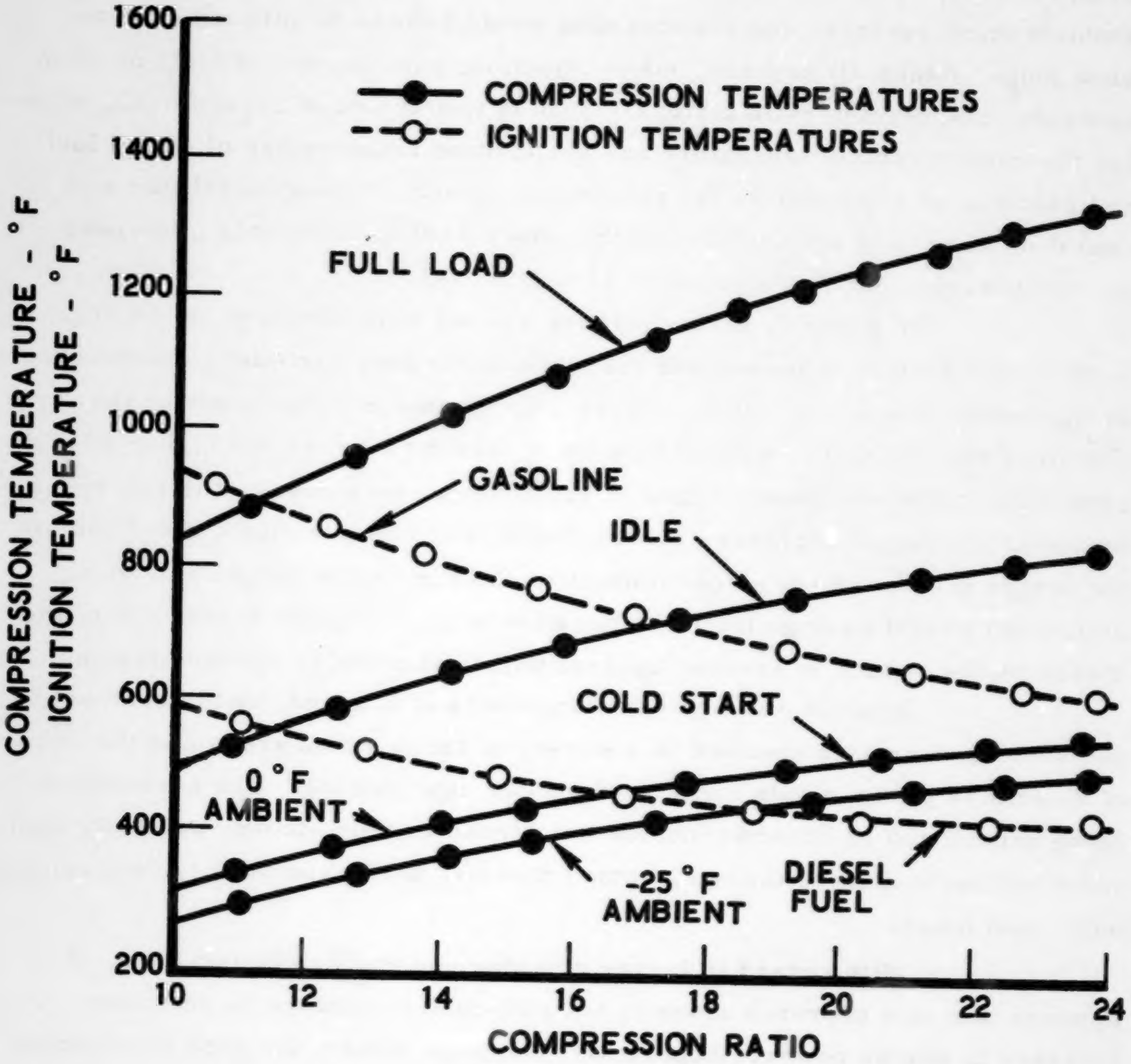


Figure 6-22. Compression and Ignition Temperatures versus Compression Ratio (Ref. 6-15)

engines are limited to a displacement of about 150 CID, and additional cylinders would be required to increase the engine displacement beyond that point. As a result, the weight and volume of the engine would increase accordingly, unless any of the above modifications were to be incorporated.

7. WANKEL AND OTHER ROTARY PISTON ENGINES

7.1 INTRODUCTION

7.1.1 General Description

In its simplest form, the Wankel engine consists of a single triangular-shaped rotor which revolves eccentrically in a chamber with a double-lobed cross section (see Figure 7-1). The action of the rotor is such that the apexes of the rotor continuously brush the walls of the chamber, thereby forming three compartments which rotate around the chamber, changing size and shape as they move by virtue of the eccentricity of the rotation. Thermodynamically, the engine is based on the Otto cycle in its "four-stroke" form. On one side of the chamber casing is a spark plug. Each compartment passes successively through all phases of the Otto cycle once in each rotor revolution. Since there are three compartments, three power strokes are produced with each rotor revolution.

7.1.2 Historical Development

Many efforts have been expended in the past century to develop internal combustion engines that will produce torque at a reasonably low speed without a crankshaft and its attendant complication. Within the last decade or so, these efforts have produced a wide variety of prototype rotary piston engines largely experimental in nature. Among these rotary engine designs the one presently receiving the greatest attention for automotive application is the Wankel Rotary Piston Engine invented by Felix Wankel in 1953. The first prototype of this engine was run successfully at the NSU Works in Germany on February 1, 1957. Later that year, the design was modified to its present basic form, the KKM-125 engine*, which is the forerunner of all current Wankel Rotary Piston Engines.

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*"KKM" stands for Kreiskolbenmotor, orbiting piston engine. The "125" designates the engine chamber volume in cubic centimeters.

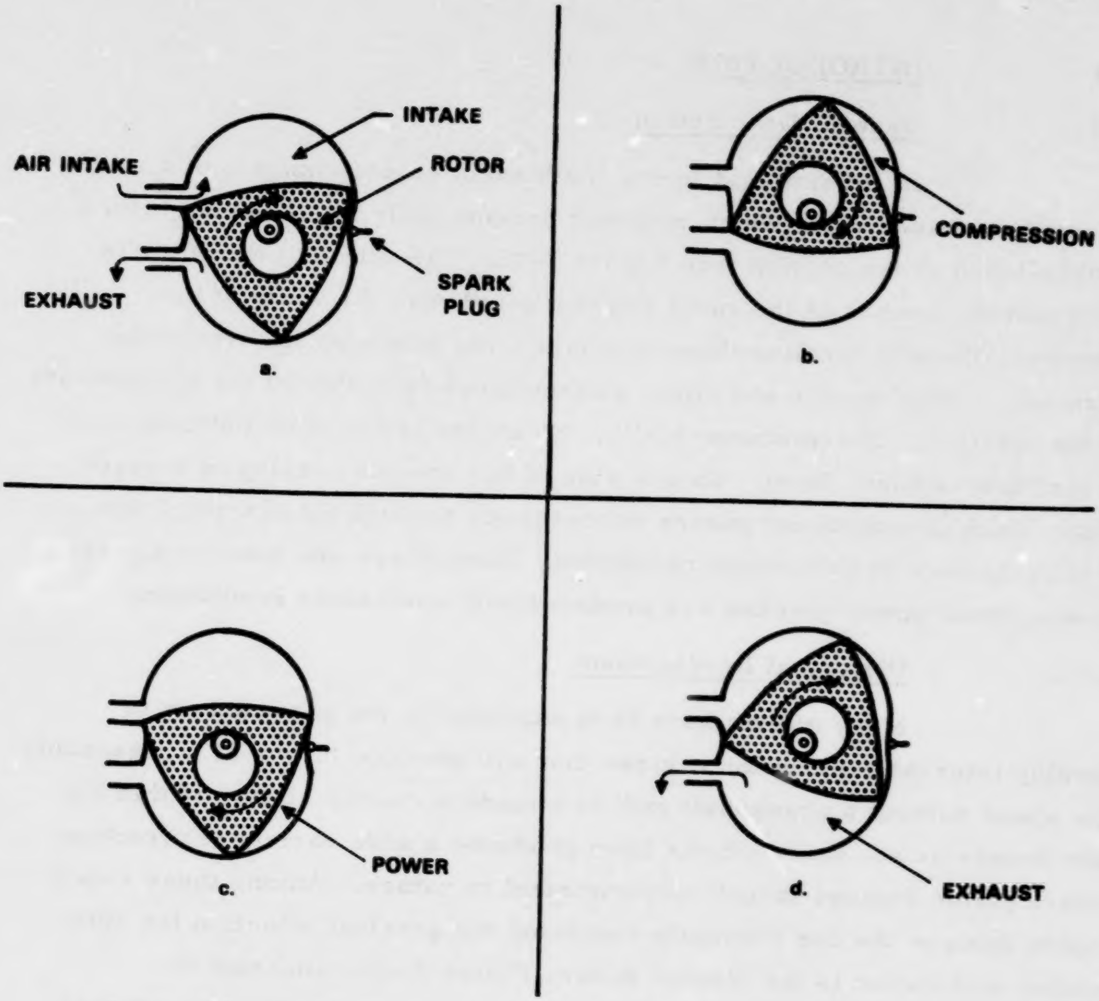

INTAKE
AIR INTAKE
ROTOR
SPARK PLUG
EXHAUST
a.
COMPRESSION
b.
POWER
c.
EXHAUST
d.

Figure 7-1. Wankel Rotary Engine

Since this early introduction, experimental engines of various sizes have been built and tested for a wide variety of applications. The patent is owned by NSU Motor Werke of Germany, which has begun to manufacture the engine in small numbers. NSU also licenses the manufacture of the Wankel engine by a number of other major companies, including Curtiss-Wright--and, more recently, General Motors--in the United States. These engines range in output power from about 0.6 hp for model airplanes to 780 hp for a Curtiss-Wright experimental engine. Typical applications cover automobiles, aircraft, boats, snowmobiles, lawn mowers, and auxiliary power units.

Current automotive rotary-piston engines are water cooled and are designed with one or two rotors for production cars and up to four rotors for experimental models. A list of vehicles equipped with these engines together with pertinent data is given in Table 7-1 (Ref. 7-1). Table 7-2 summarizes manufacturers known to be currently active in rotary engine development, together with the areas of application for the developmental engines.

7.2 POWER PLANT DESCRIPTION

7.2.1 Power Plant Configuration

Figure 7-2 shows a cutaway drawing of a typical two-rotor automotive production engine. The basic engine consists of a triangular rotor that orbits within an outer housing. The volume enclosed between the housing and each rotor flank varies as the rotor apex orbits, tracing an epitrochoid surface (the housing surface). This motion produces the four strokes of the conventional four-stroke Otto-cycle, (i.e.; intake, compression, expansion, exhaust) for each revolution of the rotor (Figure 7-3). Intake and exhaust processes occur through ports instead of valves, thus eliminating the entire valve train of the reciprocating engine which simplifies the basic engine design. The intake and exhaust flows are controlled by the

Table 7-1. Vehicles Equipped with Rotary Piston Engines (Ref. 7-1)

Vehicle	Introduction	Discontinued	Engine	Horse-power	Rotors	Compression Ratio
NSU Wankel Spider	1964	1967	NSU	64	1	8.6:1
NSU Ro-80	1967	—	NSU	136	2	9.0:1
Mazda Cosmo 110-S	1967	1969	Toyo Kogyo	110	2	9.4:1
Mazda R-100	1969	—	Toyo Kogyo	110	2	9.4:1
Mazda RX-2	1970	—	Toyo Kogyo	120	2	9.4:1
Citroen M-35	1970	Limit 500 cars	NSU	55	1	9.0:1
Mercedes Benz C-III MK I	1969	Experimental	Mercedes Benz	330	3	9.3:1
Mercedes Benz C-III MK II	1970	Experimental	Mercedes Benz	400	4	9.3:1
Mustang RC 2-60	1965	Experimental	Curtiss-Wright	185	2	8.5:1

Table 7-2. Rotary Engine Developers

Manufacturer	Approximate HP Range	Area of Application
Curtiss-Wright	up to 780	Automotive, truck, aircraft, marine, power generation, outboard motors, etc.
Mercedes Benz	up to 350	Automotive
Fichtel and Sachs	3 to 23	Lawnmowers, minibikes, snowmobiles
Graupner	0.63	Model airplanes
NSU	3-136	Automotive, portable pumps, marine
Outboard Marine	35	Marine, snowmobile
Rolls-Royce	55-180	Two-stage diesel cycle
Toyo Kogyo	90-190	Automotive
Yanmar	22-50	Outboard motors
General Motors		Automotive (1975 model year Chevrolet Vega)

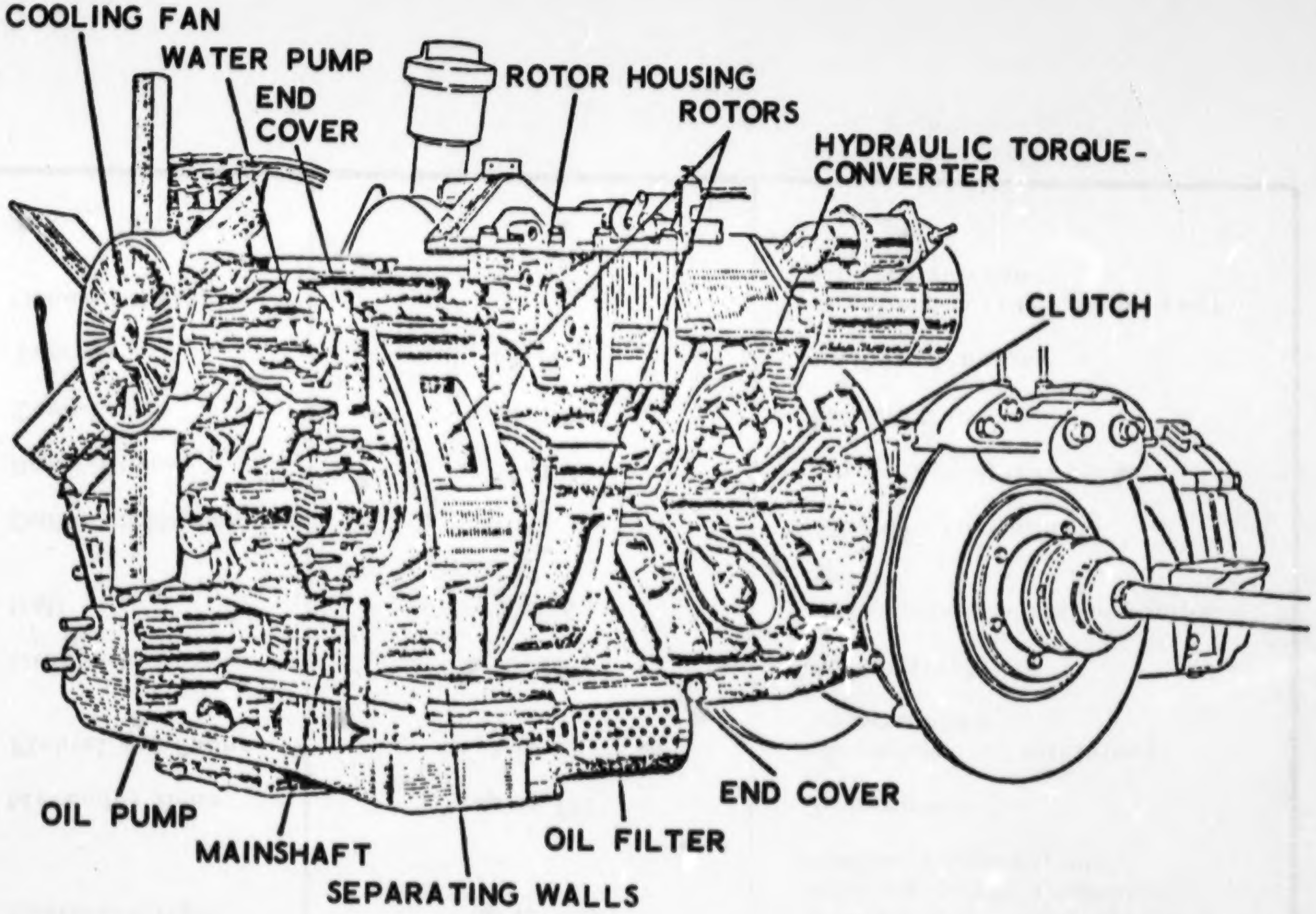

COOLING FAN
WATER PUMP
END COVER
ROTOR HOUSING
ROTORS
HYDRAULIC TORQUE-CONVERTER
CLUTCH
END COVER
OIL FILTER
SEPARATING WALLS
MAINSHAFT
OIL PUMP

Figure 7-2. Production Rotary Piston Engine for Automotive Application, Cutaway Drawing (Ref. 7-1)

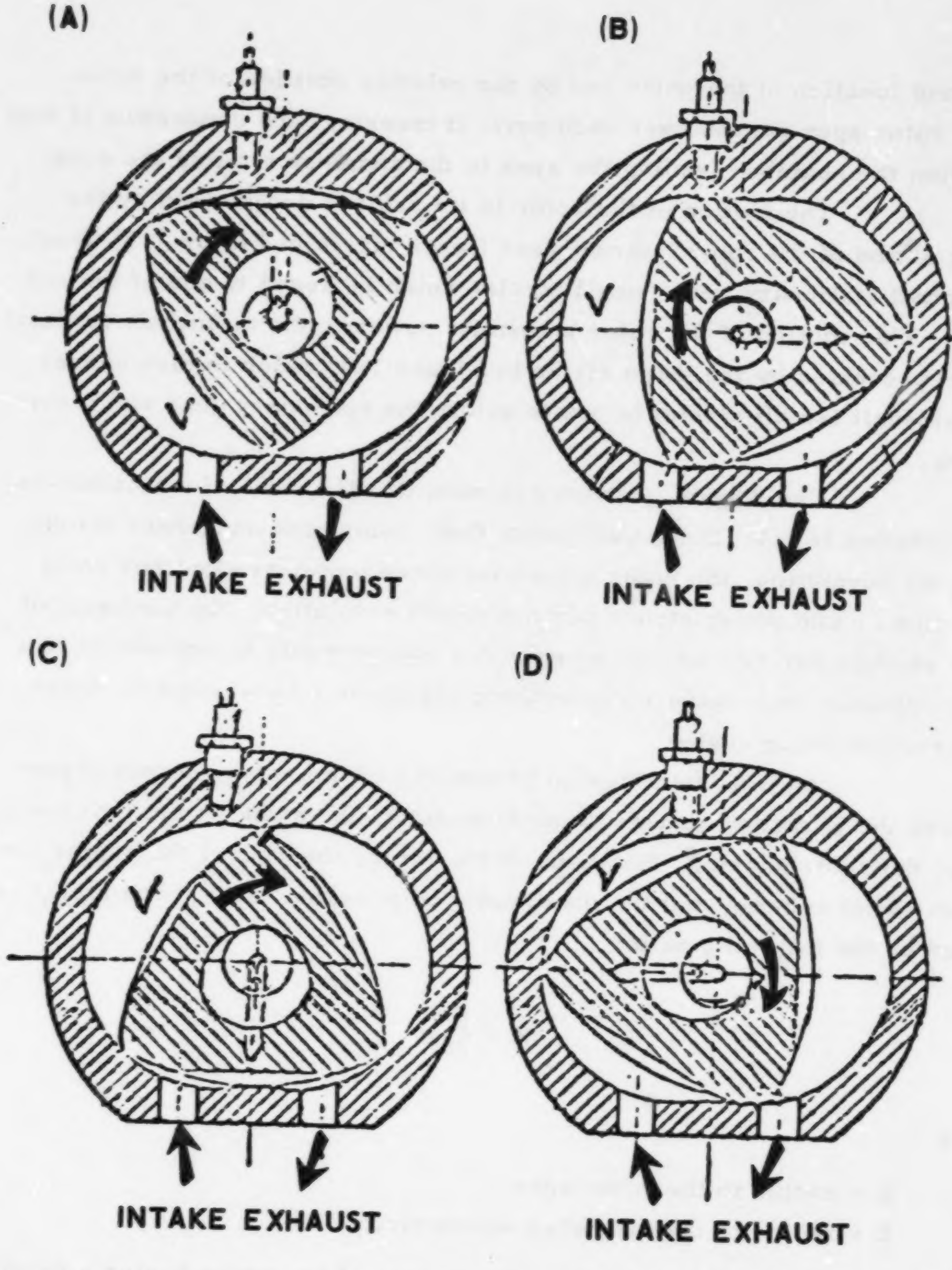

(A)
V
INTAKE EXHAUST
(B)
V
INTAKE EXHAUST
(C)
V
INTAKE EXHAUST
(D)
V
INTAKE EXHAUST

Figure 7-3. Motion of Rotor in Main Housing of Rotary Piston Engine (Ref. 7-2)

shape and location of the ports and by the relative position of the rotor. As the rotor apex passes over each port, it transfers the connection of that port from the chamber leading the apex to the chamber trailing the apex.

The motion of the rotor in its orbit is determined by the rolling action of the rotor internal gear (large circle in Figure 7-3) about the central stationary gear (small circle) which is flexed to one of the end plates. This motion of the rotor transmits power to the mainshaft eccentric (also represented by the large circle in Figure 7-3) which in turn causes the mainshaft (small circle) to rotate within the stationary gear and main bearing.

The Wankel geometry is such that the ratio of mainshaft-to-rotor rotation is 3:1. Since each rotor flank completes one power stroke per rotor revolution, the rotor completes three power strokes per rotor revolution or one power stroke per mainshaft revolution. On the basis of power strokes per revolution, a one-rotor engine would be equivalent to a single cylinder, two-cycle reciprocating engine or a two-cylinder, four-cycle reciprocating engine.

An important design parameter of the rotary-piston engine is called the K-factor, and its value determines the shape of the inner surface of the rotor housing, the shape of the rotor, the size of the engine for a given swept volume, and the maximum compression ratio. The factor is defined by the following ratio:

$$K = \frac{R}{E}$$

where

R = radius to the rotor apex
E = radius of the mainshaft eccentricity

Figure 7-4 shows the variation in relative size of an engine having a fixed swept volume and different values for the K-factor. Also shown are the maximum theoretical compression ratios for each engine.

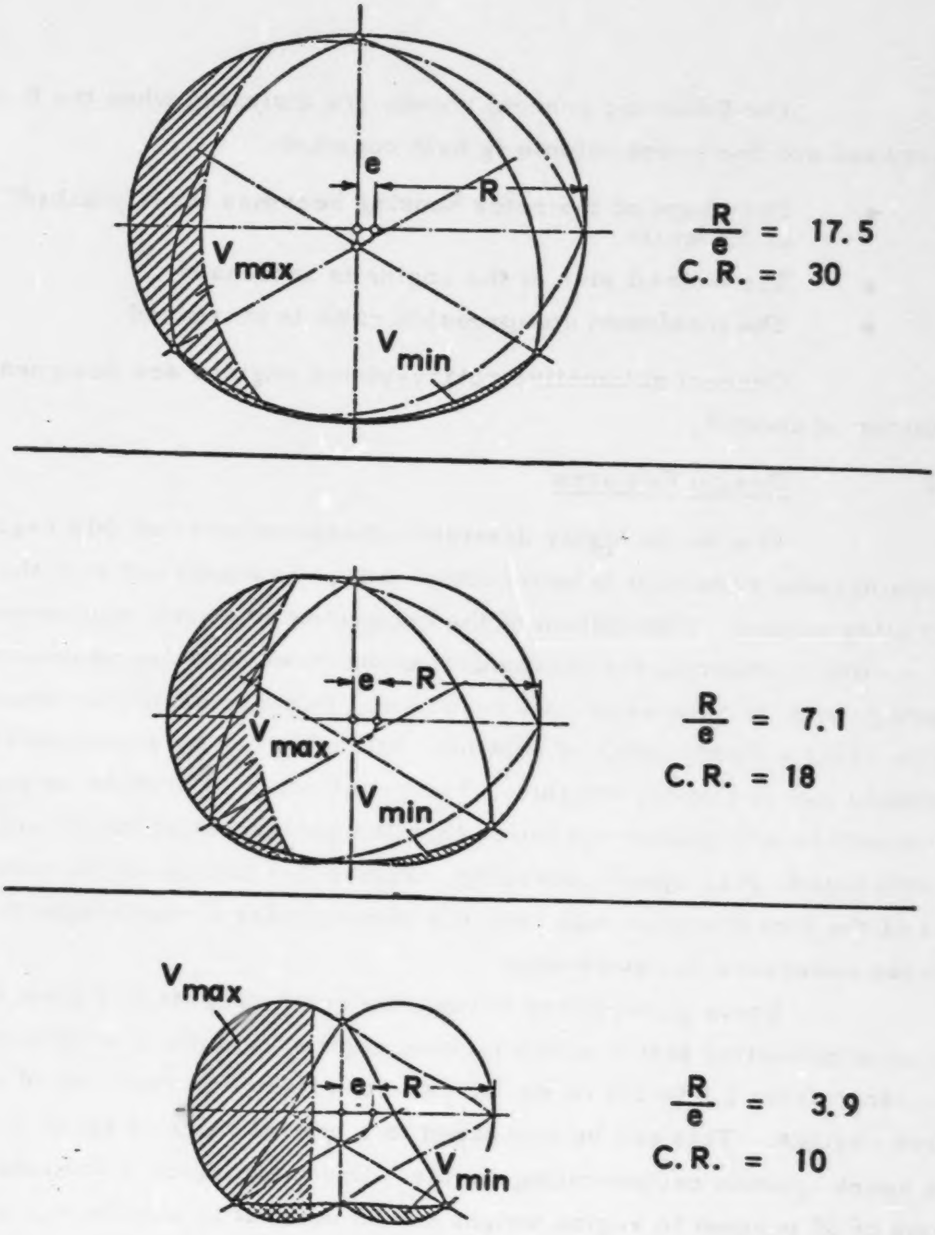

e
R
$\frac{R}{e}$ = 17.5
C.R. = 30
V_{max}
V_{min}
e
R
$\frac{R}{e}$ = 7.1
C.R. = 18
V_{max}
V_{min}
V_{max}
e
R
$\frac{R}{e}$ = 3.9
C.R. = 10
V_{min}

Figure 7-4. Variation of Size and Compression Ratio for Engines with Fixed Swept Volume and Various K-Factors (Ref. 7-1)

The following general trends are indicated when the K-factor is increased and the swept volume is held constant:

- The shape of the rotor housing becomes less "pinched" at the waist
- The overall size of the engine is increased
- The maximum compression ratio is increased.

Current automotive rotary-piston engines are designed with a K-factor of about 7.

7.2.2 Design Features

One of the highly desirable characteristics of this engine is its considerable reduction in basic complexity, as compared with the reciprocating engine. Elimination of the complete valve train and connecting rods results in lowering the mechanical noise level, greatly simplifies the balancing, and is an area of cost reduction. Because all major masses revolve about a fixed center of rotation, balancing can be accomplished by the simple use of counter weights. The net effect is to provide an engine with smoother and quieter operation than the reciprocating piston engine and with higher peak speed capability. Figure 7-5 compares the rotating parts of the rotary engine with that of a two-cylinder piston engine to illustrate the difference in complexity.

For a given power rating, the rotary engine is lighter and smaller than a reciprocating piston spark ignition engine. Values of weight-to-power ratio range from 2.0 to 2.4 lb/hp for current production versions of rotary automotive engines. This can be compared to a typical value of about 3.6 lb/hp for a spark ignition reciprocating engine. One manufacturer indicates reductions of 30 percent in engine weight and 50 percent in size from a comparable piston engine. Reductions in engine size and weight can also lead to smaller-size, lighter-weight vehicles as a result of more compact packaging. The size effect is illustrated by a profile comparison with a V-8 piston

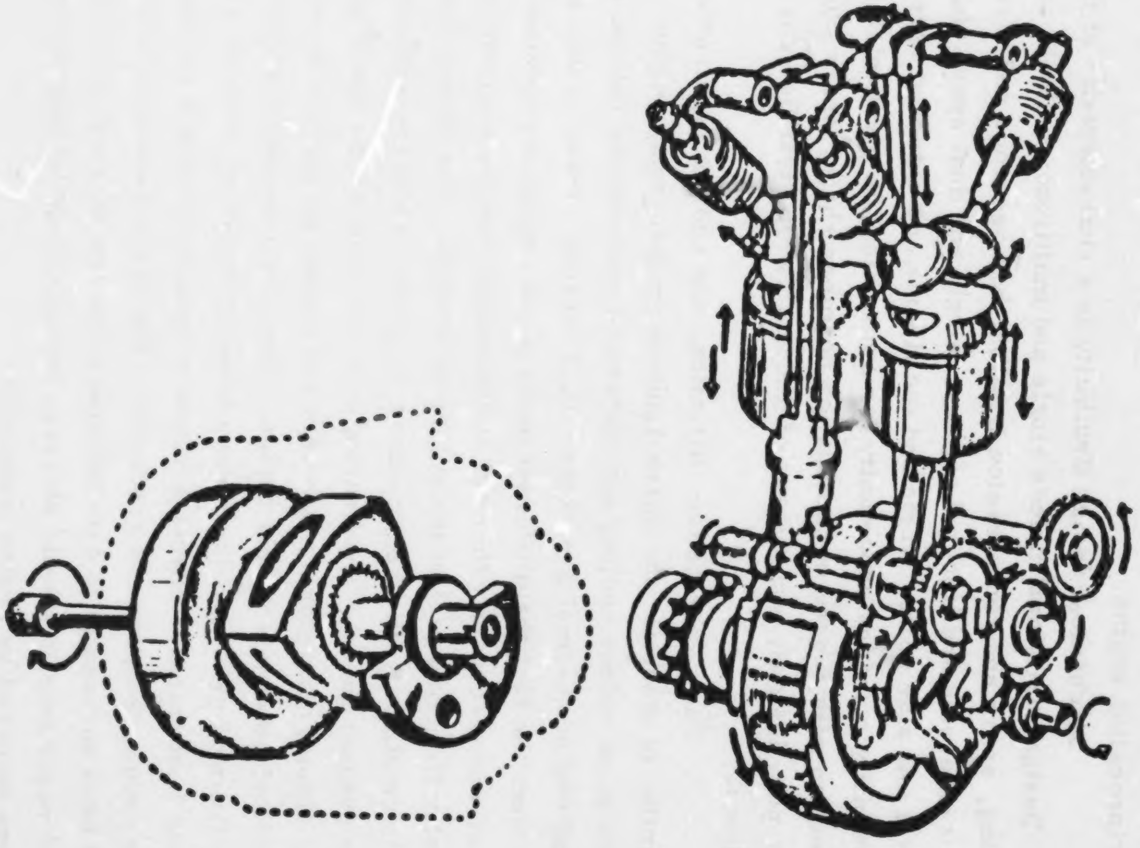


Figure 7-5. Comparison of Rotating Parts (Ref. 7-3)

engine in Figure 7-6. Figure 7-7 compares the number of parts used in the Mazda RX-2 rotary engine and the main parts of a six-cylinder, 2.7-liter reciprocating engine.

Mechanical design flexibility is a characteristic of this engine. Design variations include single and multirotors, liquid cooling and air cooling, spark and compression ignition, homogeneous and stratified charge, carbureted and injected engines, single and dual spark plugs, peripheral and side ports, single and multiports, and extended speed range. This flexibility provides opportunity for optimizing the mechanical design of this engine with regard to combustion chamber shape, spark plug location and plug-port designs, and the use of multiports for stratified charge applications.

Fuel, lubrication, filtration, and cooling requirements are very similar to those for the spark-ignition reciprocating engine. However, there are some unique cooling and material requirements related to apex seals and engine thermal gradients. High thermal stress in this engine results from the fact that different parts of the engine are exposed to conditions of a near-constant thermal environment, but the environment varies drastically from one part of the engine to another. The intake side of the engine is continually exposed to only cool gases, while the exhaust side sees only hot exhaust gases. The housing in the vicinity of the spark plug is exposed almost continuously to a very hot flame, giving rise to excessively high thermal gradients in this region. The only reasonable solution to relieve thermal gradients is to use a housing material, such as aluminum, with good heat transfer characteristics to quickly dissipate the heat and thereby reduce temperature gradients. The use of cast iron in this region has not been successful to date because of its low thermal conductivity and reduced resistance to tensile stresses brought about by high thermal gradients incurred by engine transients.

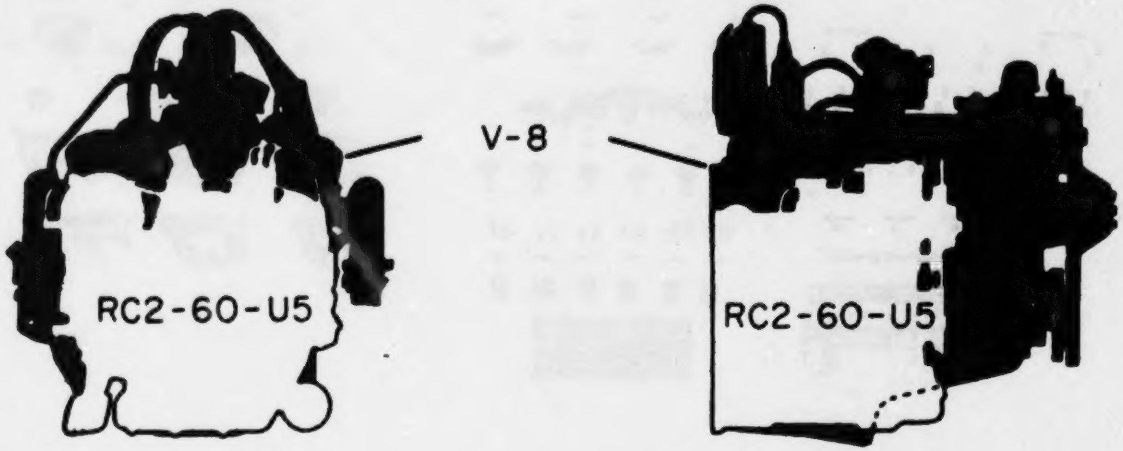

V-8
RC2-60-U5
RC2-60-U5

Figure 7-6. Silhouette Comparison of a Twin-Rotor Wankel Engine (Curtiss-Wright RC 2-60 U5) with a 283-cubic inch Chevrolet V-8 Piston Engine (Ref. 7-1)

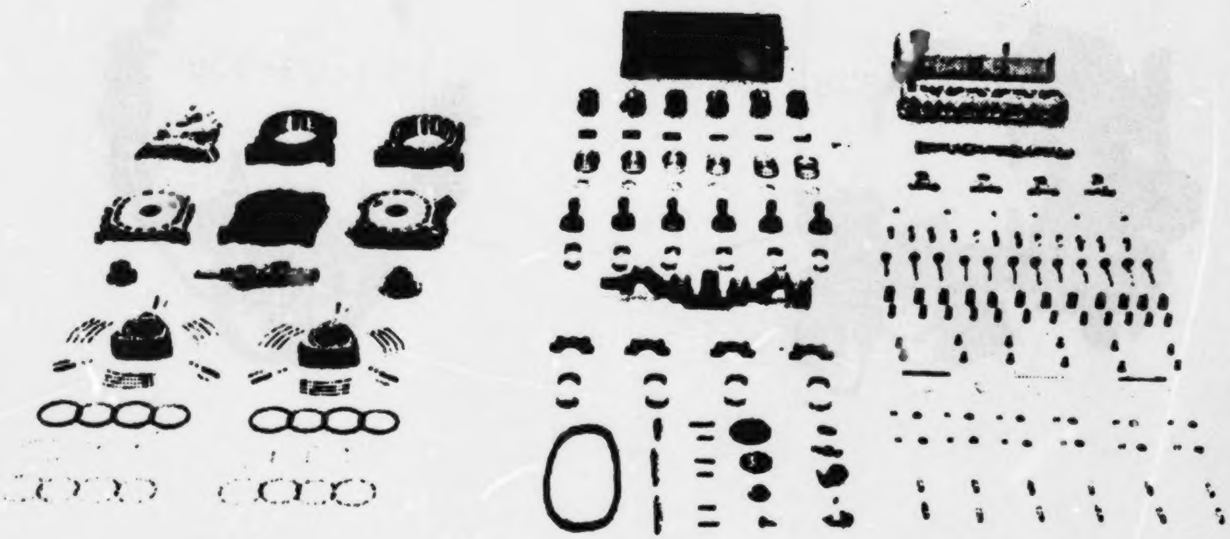


Figure 7-7. Comparison of Main Parts for RX-2 Rotary Engine and 6-Cylinder Reciprocating Engine (Ref. 7-4)

7.2.3 Operating Characteristics

Engine operating characteristics are essentially identical to those for the spark-ignition, reciprocating piston engine, and the rotary engine can be linked to conventional transmissions. Transmission-gear ratios may be revised somewhat to compensate for the fact that the rotary engine achieves high torque at higher peak speeds, similar in this characteristic to some small, four-cylinder reciprocating engines.

7.3 PERFORMANCE CHARACTERISTICS

The basic engine produces high levels of hydrocarbon and carbon monoxide exhaust emissions. These result from seal leakage, wall quenching from high surface-to-volume ratios and crevices, and incomplete combustion (Ref. 7-5). The same factors contribute to low combustion temperatures and resulting low levels of NO_x.

The control of HC and CO emissions in current (1973) production rotary engines is accomplished by the use of air injection into a thermal reactor coupled to the exhaust ports. Typical results achieved on the 1973 model year Mazda (RX-3) are shown in Table 7-3, together with results of development tests performed by Toyo Kogyo and General Motors with the objective of meeting the original 1975 standards with the thermal reactor system (Ref. 7-6).

The best results achieved with oxidation catalysts are shown in Table 7-4. Results at low mileage are seen to approach the 1976 standards but do not meet the 1977 NO_x standards (Ref. 7-6). It should be noted that General Motors (Ref. 7-7), Ford (Ref. 7-8), and Daimler-Benz (Ref. 7-9) have all reported extremely poor durability results to date for oxidation catalyst systems used with rotary engines.

The fuel economy of 1973 model year rotary engine cars (i.e., Mazda) is approximately 30 percent below piston engine vehicles of the same inertia weight (Ref. 7-10). Development work reported by General

Table 7-3. Best Emissions Results, Thermal Reactor and Wankel Engine, 2,750-lb Compact Car (Ref. 7-6)

Manufacturer	System*	Mileage	No. of Tests	Emissions in gm/mi **			Fuel Penalty***
				HC	CO	NO_x	percent
Toyo Kogyo (Mazda)	Reactor		1973 Certification	2.4	20.0	0.9	0 (Baseline)
Toyo Kogyo	Reactor	4,000	Many	0.32	3.1	0.83	6.5
		50,000	3	0.36	2.6	0.87	--
	Reactor + EGR	Low	8	0.35	2.2	0.49	12.0
General Motors	Reactor	Low	72	0.60	5.0	0.60	--
	(best effort)	Low	1	0.43	2.8	0.44	--

*All these systems are carbureted fuel rich (about 12:1)

**1975 CVS-CH test procedure

***Relative to 1973 production rotary engine car

Table 7-4. Emissions at Low-Mileage Rotary Engine with Oxidation Catalyst* (Ref. 7-6)

System	No. of Tests	Emissions, grams/mile**		
		HC	CO	NO_x
Oxidation catalyst and air pump	22	0.7	0.4	1.2
(Best Effort)	1	0.4	0.2	1.0

*General Motors data, 2750-lb car

**1975 CVS-CH test procedure

Motors (Ref. 7-7), on a modified Mazda engine (lean carburetor, modified ignition timing) tailored for fuel economy without emission control devices, compared favorably to a 1971 Opel 1.9-liter vehicle at 24 to 29 mpg.

Leakage of gases from the combustion chamber is one of the more perplexing problems introduced by the rotary engine. This leakage results primarily from the necessary use of an apex seal having a single line of contact with the outer housing; it has a strong effect on both fuel economy and exhaust emissions. A view of a typical apex seal is given in Figure 7-8.

Current apex seal materials appear to have a reasonable wear life of about 80,000 miles. However, even with new seals, gas leakage is excessive. This appears to be due in part to the hydrodynamic effect resulting in seal lift from the housing surface and the subsequent leakage of high pressure combustion gases into both the leading and trailing chambers. Gases leaking into the leading chamber have a high concentration of unburned

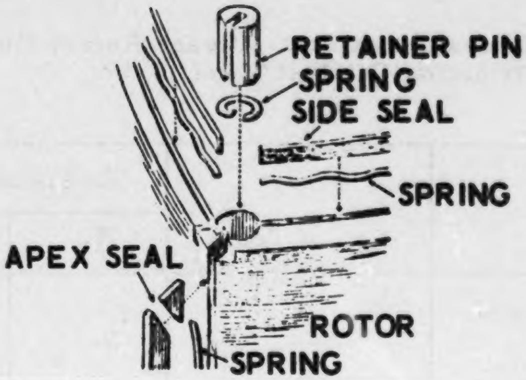


Figure 7-8. Typical Apex Seal (Ref. 7-1)

mixture, leading to loss of power (poor fuel economy) and to high hydrocarbon concentrations in the exhaust. A lowering of combustion chamber peak pressure also results.

Improvements in sealing will reduce the mixture loss to the leading chamber, thus improving fuel economy and decreasing hydrocarbon emissions; but this change will also increase peak combustion pressures, thereby increasing the production of NO_x. Therefore, it can be expected that improved sealing will tend to correct the fuel economy and hydrocarbon emission problems, but will contribute to the NO_x emission problem.

7.4 CURRENT STATUS OF TECHNOLOGY

Rotary engine-powered cars constitute a very small fraction of the current automobile population, but production levels have been rising. Toyo Kogyo is presently the primary producer of these engines, but they may be surpassed by General Motors if the Chevrolet Vega rotary engine is produced in model year 1975, as scheduled, and finds large public acceptance.

Available information indicates a lower rate of production (engines/hour/line) as compared with the piston engine. This results from the complex machining operations that are required to grind and finish the working surfaces of both the housing and rotor. Published information (Ref. 7-11) on production machines give capabilities of around 25 housings per hour per line as compared with typical production rates for piston engines of over 100 per hour. This limitation appears to have a significant impact on manufacturing costs. Either a major breakthrough in the manufacturing process or the use of parallel production lines will be required to achieve current piston-engine production rates.

7.5 PROJECTED STATUS

The more important problem areas facing the rotary-engine designer are:

- Gas seal leakage
- Low speed fuel economy
- High level of exhaust emissions
- High thermal stresses.

Low fuel economy, especially at low speeds, is related to the leakage problem and to the characteristic slow burning rate of the rotary engine. A solution to the sealing problem will help improve fuel economy. The slow burning rate, as evidenced in pressure-volume plots, shows a drastic departure from the ideal constant-volume combustion of the Otto Cycle, resulting in a drop in efficiency. The use of dual ignition ports could produce some increase in the burning rate, which may improve combustion characteristics.

The high surface-to-volume ratio is an inherent characteristic of this engine and cannot be modified to any great extent. Crevice effects can be reduced down to some minimum by proper design, but they cannot be eliminated. Seal leakage has already been discussed and potentially large improvements can be made here, while incomplete combustion can be improved with further research leading to a better understanding of the unique combustion process in the rotary engine. Stratified charge combustion is a strong possibility in this engine, which appears to offer lower emissions and higher fuel economy (Ref. 7-12).

Further optimization in rotary engine design, including the use of stratified charge combustion and improved versions of apex seals and thermal reactors, may overcome current fuel economy and emissions problems of the basic engine. (With the addition of a thermal reactor, the Mazda rotary engine has demonstrated the capability of meeting the original 1975 Federal emission standards.)

8. STRATIFIED CHARGE ENGINES

8.1 INTRODUCTION

8.1.1 General Description

In principle, the stratified charge engine represents a modification of the conventional spark ignition engine. The main objective of the modification is to achieve heterogeneous combustion in which a rich fuel mixture is generated around the spark plug and a lean mixture is generated in remaining combustion chamber zones. The resulting two-stage combustion process permits operation of the engine at very lean overall fuel-air ratios which is conducive to low emissions, good fuel economy, and reduced sensitivity of the engine to fuel octane and cetane numbers.

Stratified charge engines can be divided into two distinct classes: open chamber and divided chamber. In open-chamber configurations, exemplified by the Texaco TCCS and Ford PROCO engines, a single combustion chamber similar to that of conventional spark ignition engines is used. During engine operation, an air swirl is set up in the cylinders by means of directional intake porting combined with special piston cup designs. Fuel is injected into each cylinder toward the end of the compression stroke. Upon ignition of the swirling rich mixture surrounding the spark plug, the burning charge expands into the combustion chamber's outer regions where the combustion process is completed in an oxygen-rich environment. Attempts are currently being made to replace the more expensive fuel injection systems employed in these engines with conventional carburetors.

The divided-chamber stratified charge engines or prechamber engines, exemplified by Honda's CVCC engine concept, employ two interconnected combustion chambers per cylinder. During the induction stroke of the piston, a fuel-rich mixture is inducted into the generally smaller prechamber while the main chamber is charged with a lean mixture or even pure air.

Upon ignition in the prechamber, hot gases expand into the main chamber where combustion is then carried to completion. The principal advantage of prechamber engines over conventional engines is their ability to operate with very lean overall fuel air mixtures resulting in low emissions, particularly NO_x. However, because of the less favorable combustion chamber surface-to-volume ratio combined with high turbulence, the heat losses of this engine tend to be higher than in conventional designs. The benefits in terms of emission reduction and fuel economy improvement that might be realized in a particular design depend upon the tradeoffs between the heat losses and the inherently higher thermodynamic cycle efficiency obtained with operation in the lean fuel air mixture regime.

8.1.2 Historical Developments

The concept of stratified charge operation of internal combustion engines and its potential was first evaluated in the early 1920s by Ricardo in England. In the United States, work on the open-chamber configurations commenced in the 1940s with Texaco's invention of a "knockless" engine. Research and development of a number of alternate designs have since been pursued by several investigators and organizations throughout the world. In 1965, the United States Army Tank Automotive Command (TACOM) initiated funding of exploratory research of both open-chamber and divided-chamber stratified charge engine concepts which culminated in the development efforts conducted over a number of years by Ford and Texaco. The objective of the TACOM-sponsored Ford and Texaco programs was the development of a military engine with multifuel capability and improved fuel economy relative to conventional spark ignition engines. Minimization of exhaust emissions was added later as a design requirement under joint EPA and TACOM sponsorship. The level of funding of the Texaco effort had been recently increased prior to completion of the program. The TACOM support of the Ford development program was previously terminated. However, Ford is conducting a number of in-house programs aimed at the development of its stratified charge engine concepts for potential use in passenger cars.

The concept of a divided combustion chamber can be traced back to the first "oil engines" in the pre-Diesel era. These early prechamber engines operated on naphtha or heavy oils that were injected into a spherical prechamber and vaporized in contact with the red-hot thermally insulated prechamber walls. Later, attempts were made to apply the prechamber concept to the spark ignition gasoline engine, as documented by numerous patents. However, to date, a practical and widespread application of the prechamber concept has been realized only in high-speed diesel engines, principally due to the pioneering efforts of Ricardo. Currently, a number of organizations and individuals are working on the development of prechamber engines. In particular, the Honda Motor Company of Japan has conducted extensive design, development, and test work on its CVCC concept during the past several years.

8.2 OPEN-CHAMBER STRATIFIED CHARGE ENGINES

8.2.1 General

Since, as previously noted, Ricardo developed the engineering concept of the stratified charge operation (early 1920s), and since Texaco's invention of the "knockless engine" in the 1940s, many investigators have conducted research on a number of alternate designs throughout the world. In 1965 the U.S. Army Tank Automotive Command (USATACOM) started to fund exploratory research on various stratified charge engine concepts in an effort to develop an engine with multifuel capability and better fuel economy than conventional spark ignition engines. These engine designs included the Texaco Combustion Process, the Borg Warner Combustion System, the Witzky Swirl Stratification System, and two divided chamber configurations then under development at Continental Aviation and Engineering and at the University of Rochester. Based on this initial work, TACOM subsequently selected the Ford Combustion Process (FCP) and the Texaco Combustion Process (TCP) for further development and evaluation (Ref. 8-1). Low exhaust emissions were added later as a design requirement under joint TACOM and EPA funding.

8.2.2 Power Plant Description

A number of the open chamber, stratified charge engine concepts developed to date are described in the following sections. These include the Baudry engine, the Borg Warner Combustion System, the Hesselmann engine, the Mitsubishi Combustion Process, the Witzky engine, the Texaco Controlled Combustion System, and the Ford Combustion Process.

8.2.2.1 Baudry Engine

This engine concept, illustrated in Figure 8-1, was developed by the French Petroleum Institute (Ref. 8-2). As shown in the figure, a small tube provides a fuel-rich mixture at the cylinder inlet. This mixture is then carried by the in-rushing air towards the spark plug which is located at the periphery of the combustion chamber. Although the operation of the engine is difficult to explain, Baudry reports remarkable improvements in part-load fuel economy and CO emissions relative to conventional spark ignition engines, particularly in the low power regime.

8.2.2.2 Borg Warner Combustion System

The Borg Warner Combustion System is shown in Figure 8-2. In this concept, charge stratification is achieved by injecting the fuel into the piston cavity at bottom dead center. The fuel-rich combustion mixture is contained in the cavity during the compression stroke, and combustion is initiated by means of a spark plug near piston top dead center (Ref. 8-1).

8.2.2.3 Hesselmann Engine

The Hesselmann engine, shown schematically in Figure 8-3, was designed in the early 1930s with the intention to combine the advantages of the Diesel and Otto cycle engines. This engine, which is also known as the Hesselmann "low-pressure diesel," utilizes a high air swirl in the combustion chamber, generated by shrouded intake valves. The fuel is injected towards the center of the cylinder through a nozzle positioned at the periphery

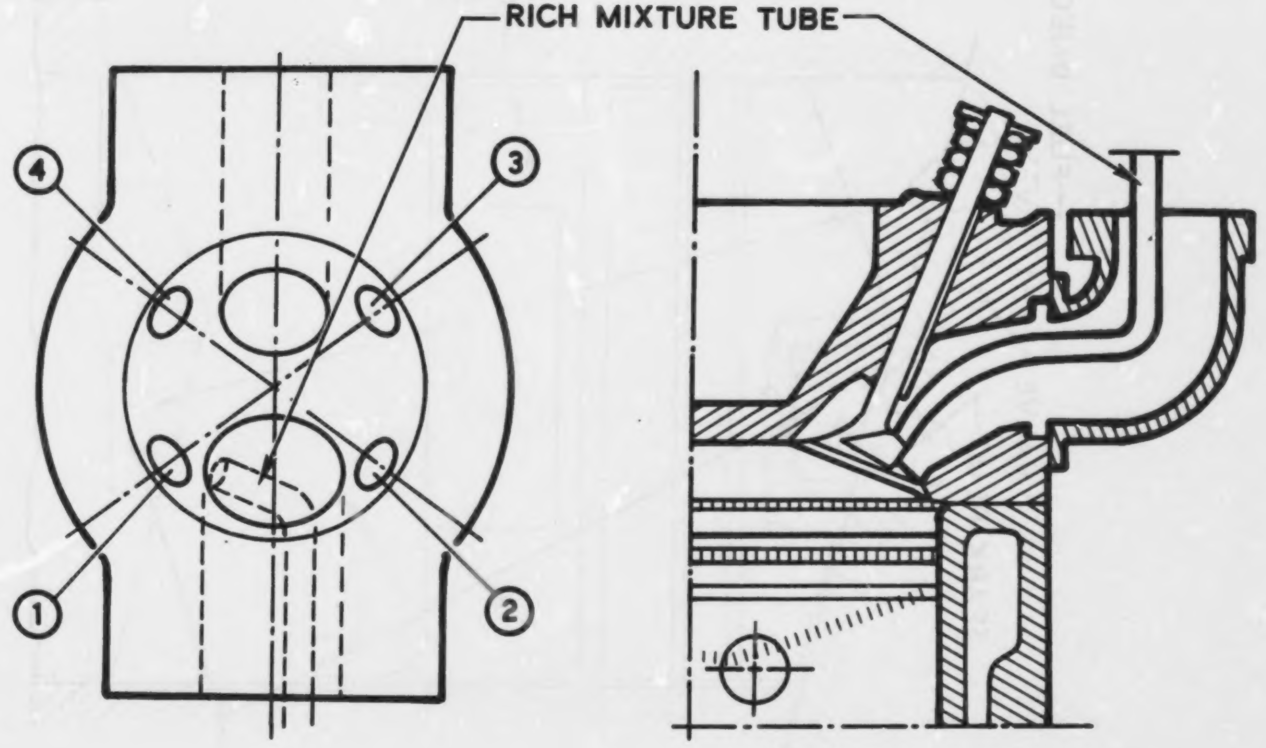

RICH MIXTURE TUBE
4
3
1
2

Figure 8-1. Baudry Stratified Charge Engine (Ref. 8-2)

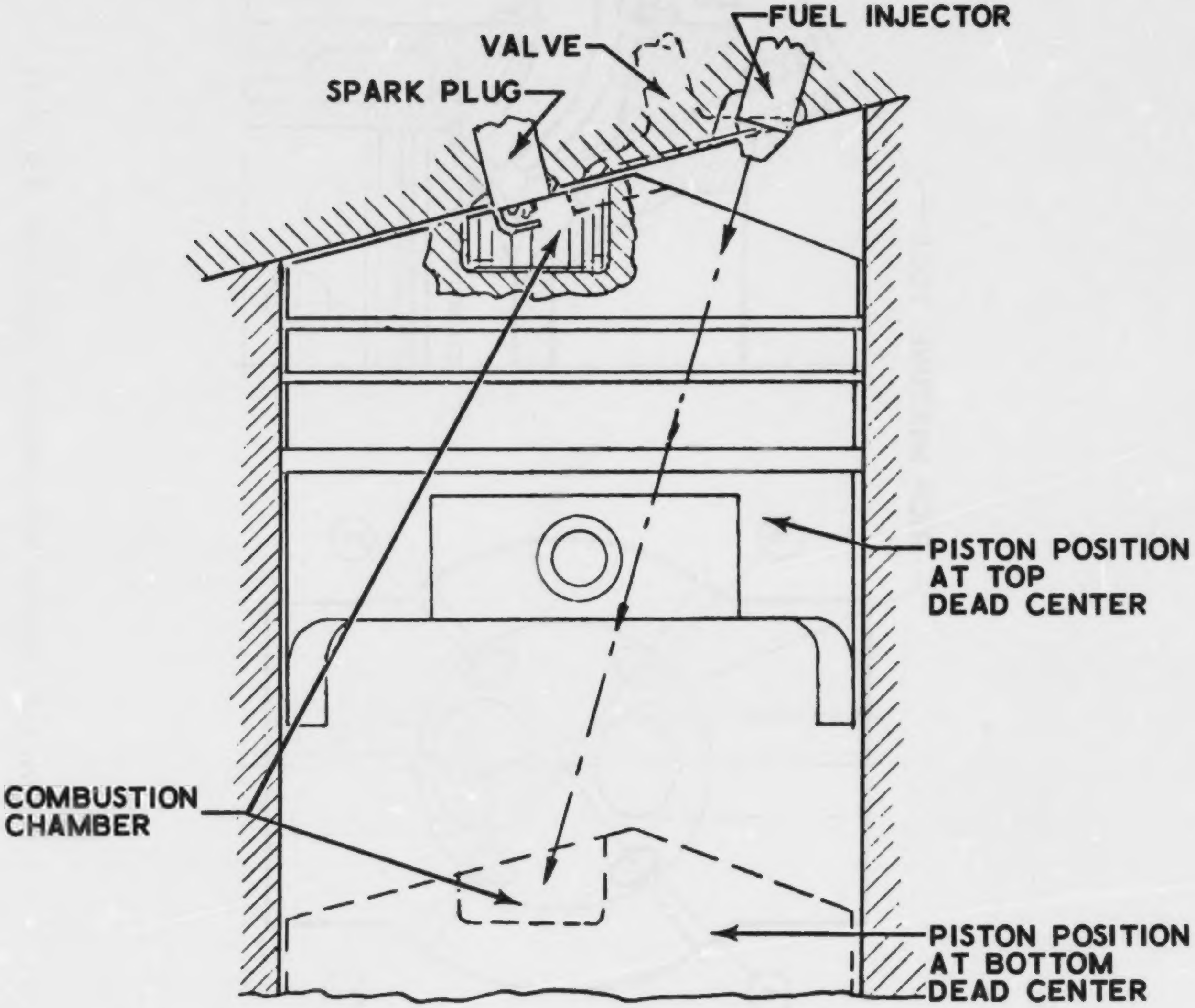

FUEL INJECTOR
VALVE
SPARK PLUG
PISTON POSITION
AT TOP
DEAD CENTER
COMBUSTION
CHAMBER
PISTON POSITION
AT BOTTOM
DEAD CENTER

Figure 8-2. Borg Warner Combustion System (Ref. 8-1)

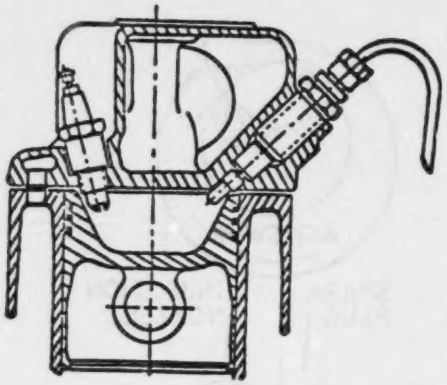


Figure 8-3. Hesselmann Stratified Charge Engine (Ref. 8-3)

of the combustion chamber. Fuel injection occurs during the compression stroke of the piston. Ignition of the combustible fuel-air mixture is accomplished after completion of the injection process by means of a spark plug located diametrically opposite the injection nozzle. Except at idle, the engine operates unthrottled, and its power output is controlled by the fuel supply (Ref. 8-3).

8.2.2.4 Mitsubishi Combustion Process

The Mitsubishi Combustion Process (MCP) is illustrated in Figure 8-4 (Ref. 8-4). In this engine, an air swirl is generated by spiral induction ports, and the fuel is injected against this swirl into the piston cavity. Tests conducted by Mitsubishi indicate low emission and fuel consumption characteristics, especially at light loads. Also, the engine has been run successfully on a variety of fuels including gasoline, kerosene, fuel oil, and diesel fuel. The engine exhibits good starting characteristics, even at temperatures as low as -22°F, and its noise and vibration levels are comparable to conventional spark ignition engines (Ref. 8-4).

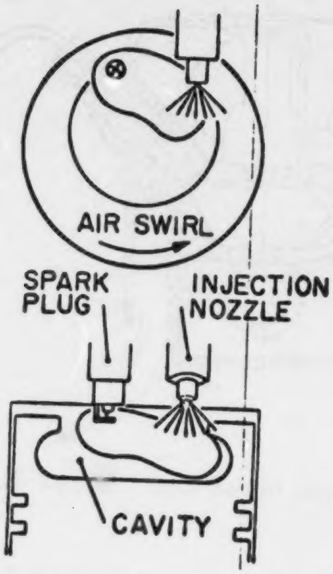


Figure 8-4. Mitsubishi Combustion Process (Ref. 8-4)

8.2.2.5 Witzky Engine

This engine concept is depicted in Figure 8-5. It is based on the unthrottled Otto cycle process and utilizes direct fuel injection into the cylinder during the last portion of the compression stroke. The fuel is injected against an air swirl, set up by special intake ports, in such a manner that a fuel-rich mixture is formed in the vicinity of a centrally located spark plug (Ref 8-5). Upon ignition in the rich zone, the flame spreads into the lean outer regions of the combustion chamber. The power output of the engine is controlled by varying the amount of fuel injected. According to Witzky, the engine has been successfully operated at very lean air-fuel mixtures and with low octane fuels. Other advantages are claimed to include smooth combustion with low peak pressures and pressure rise rates, good part-load fuel economy, low combustion noise, good acceleration characteristics, and ease of starting even at low ambient temperatures (Ref. 8-3).

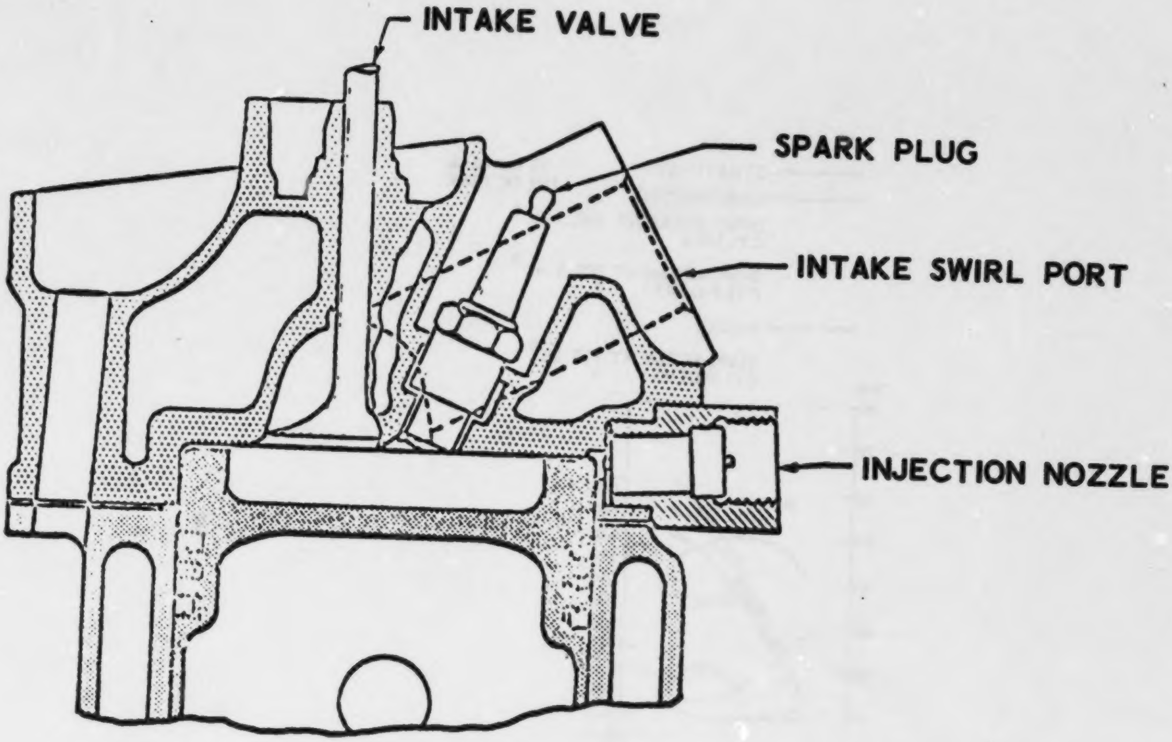


Figure 8-5. Witzky Stratified Charge Engine (Ref. 8-5)

The Witzky system has been incorporated by the Southwest Research Institute into a 215-CID Buick aluminum V-8 engine (Ref. 8-3). In this installation, every other cylinder in the firing order was converted to stratified charge operation, while the remaining four cylinders remained unmodified. Figure 8-6 shows the measured full-throttle performance of the "four-cylinder" stratified charge engine adjusted to account for the friction losses of the four nonoperative cylinders. Also shown in this figure are performance curves for the "four-cylinder" carbureted baseline engine and for the Mercedes Benz 190D light-duty automobile diesel engine. As indicated, the specific fuel consumption of the Witzky stratified charge engine is comparable to that of the diesel and substantially below the baseline engine, particularly at low speeds.

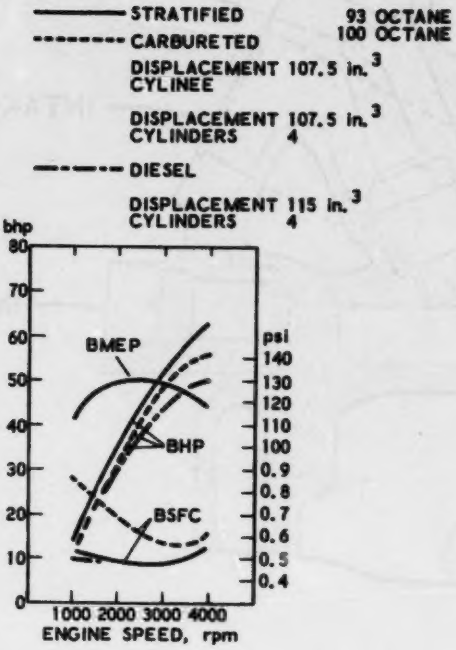


Figure 8-6. Full Throttle Performance of Witzky Stratified Charge Engine, Conventional Gasoline Engine, and Diesel Engine (Ref. 8-3)

8.2.2.6 Texaco Controlled Combustion System

8.2.2.6.1 Concept Description

The Texaco Controlled Combustion System (TCCS), illustrated in Figures 8-7 and 8-8, is a stratified charge process utilizing controlled air swirl and fuel injection combined with positive spark ignition (Refs. 8-6 through 8-8). Injection of the fuel into the swirling air commences near the end of the compression stroke, and the injected fuel is immediately ignited to establish a flame front adjacent to the injection nozzle. As injection continues, additional fuel is supplied to the flame front and burned very rapidly as soon as a combustible mixture has been formed.

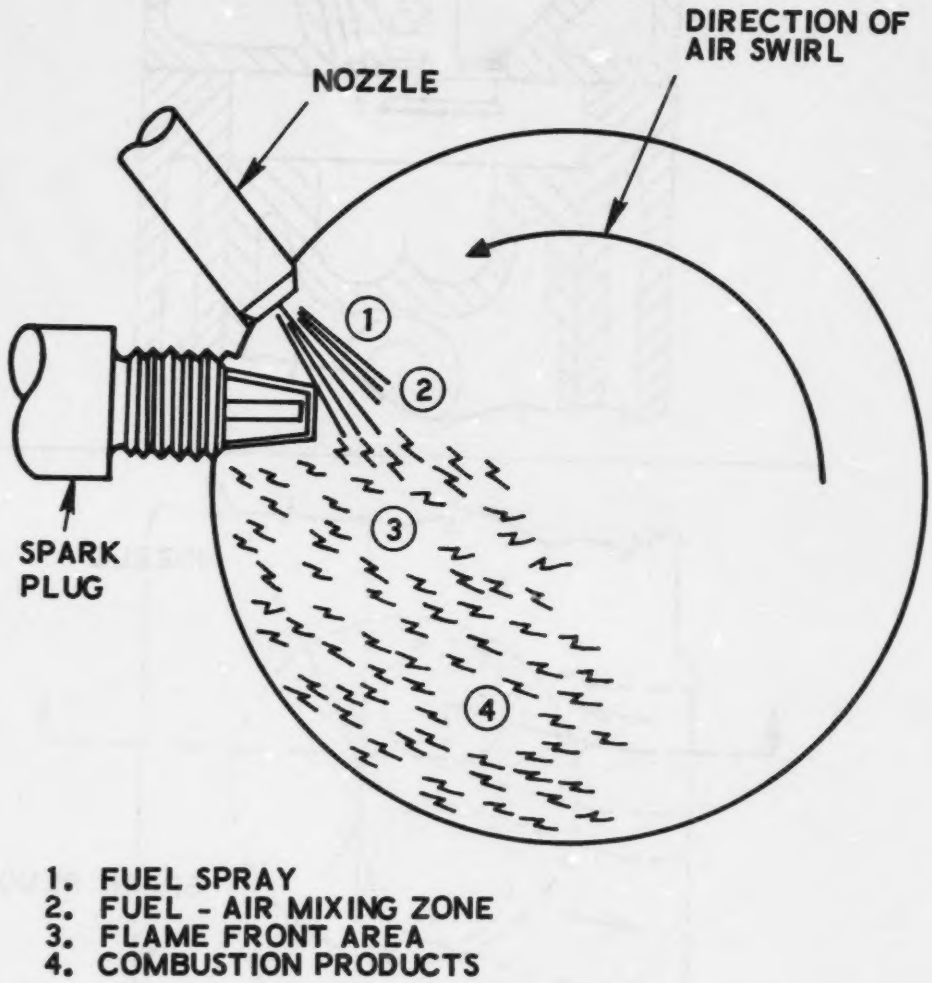

DIRECTION OF AIR SWIRL
NOZZLE
①
②
③
④
SPARK PLUG
1. FUEL SPRAY
2. FUEL - AIR MIXING ZONE
3. FLAME FRONT AREA
4. COMBUSTION PRODUCTS

Figure 8-7. Texaco Controlled Combustion System

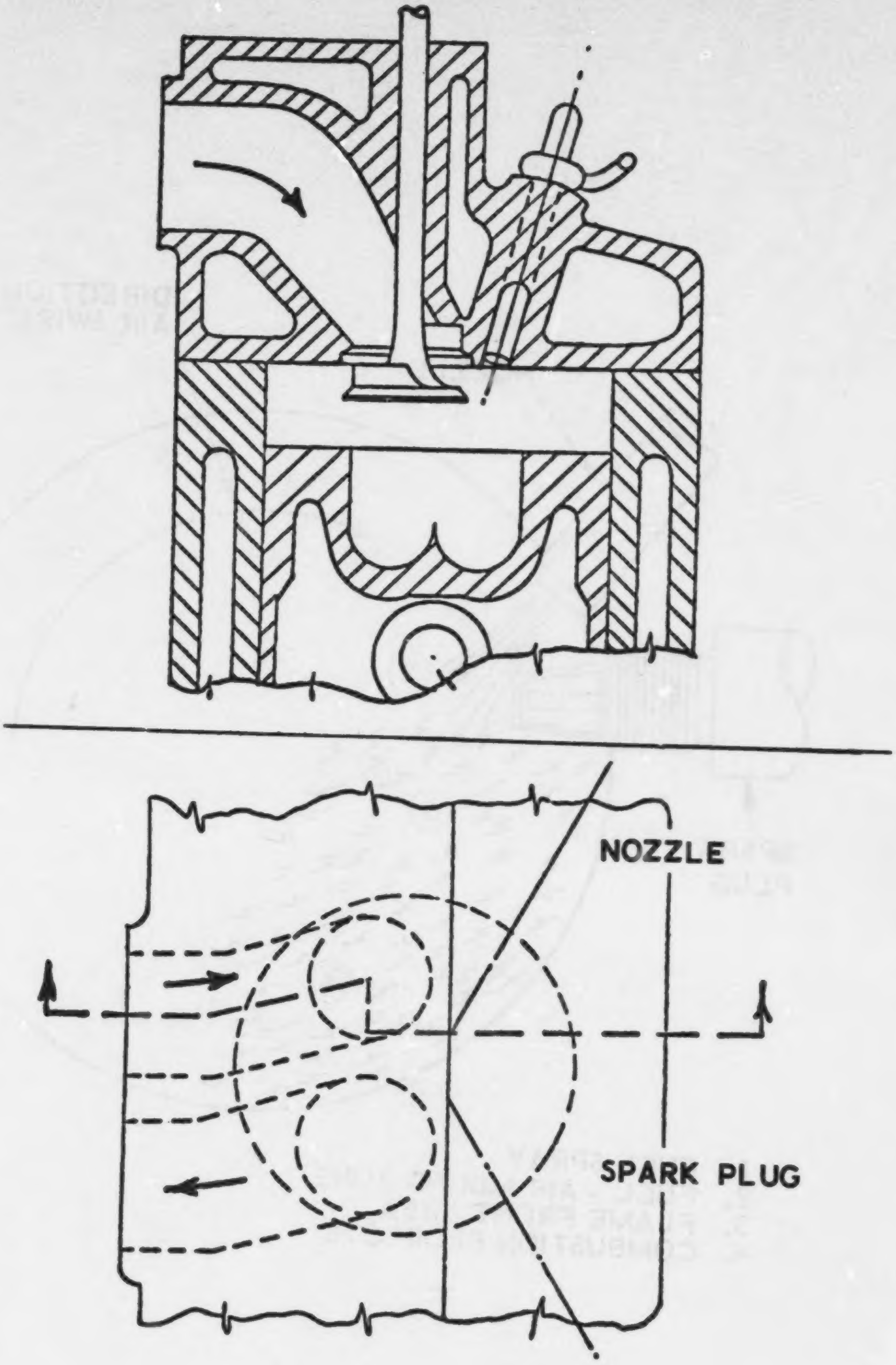

NOZZLE
SPARK PLUG

Figure 8-8. Texaco Cup Combustion Chamber, Stratified Charge Engine (Ref. 8-8)

The TCCS engine is designed to operate unthrottled with the power output being controlled by the duration and amount of fuel injected into the combustion chamber. At full load, the injection duration covers about one revolution of the air swirl. Lean fuel air mixtures are utilized at part-load and near-stoichiometric mixtures are used at full load. Because of the lean mixture operation, the engine exhibits excellent part-load efficiencies. However, at full load, the fuel air mixing process in the TCCS engine is not as efficient as in conventional spark ignition engines and, as a result, the maximum power output of TCCS is smoke limited. Owing to the direct cylinder injection and positive ignition, the engine has excellent throttle response and very good starting and warmup characteristics.

Since the TCCS engine utilizes controlled air swirl, fuel injection, and positive ignition, it is not penalized by some of the inherent disadvantages of conventional spark ignition and diesel engines. Ignition of the first increment of fuel at the time of injection eliminates the possibility of mixture preignition and engine knock frequently encountered in gasoline engines, regardless of the octane number of the fuel used. Relative to diesels, the positive ignition utilized in the TCCS engine minimizes the ignition delay and the sensitivity of the engine to fuel cetane number. Since the ignition delay is very short, the peak cylinder pressure and the pressure rise rates are considerably lower than in diesel engines. Therefore, the mechanical loads and shocks occurring during combustion are less severe, permitting the use of lightweight engine blocks, cylinder heads, crankshafts, and pistons.

Work on the TCCS concept has been under way at Texaco since the early 1940s. Many single-cylinder and multicylinder configurations have since been evaluated for potential application in industrial and automotive installations. Since 1965, Texaco has been under contract to TACOM to adapt the TCCS concept to the existing military light-duty L-141 engine which is used in the M-151 jeep vehicle. The principal design objectives established by TACOM include good fuel economy, broad fuel tolerance, and good driveability.

Both naturally-aspirated and turbocharged TCCS engines have been evaluated by Texaco. The design features of these two concepts are described in the following sections.

8.2.2.6.2 Naturally Aspirated L-141 TCCS Engine

The L-141 engine is a water-cooled, four-cylinder, in-line spark ignition engine having a cylinder bore of 3.875 inches, a stroke of 3.00 inches, and a total displacement of 141.5 cubic inches. The specifications of the standard L-141 engine and the TCCS modification are listed in Table 8-1.

Because of the oversquare bore-to-stroke ratio of the engine, incorporation of a cup-in-piston combustion chamber is required to ensure adequate swirl rates during combustion. As shown in Figure 8-8, the piston cup has a cylindrical upper portion and a toroidal bottom. The cup diameter of about one-half of the bore has been determined by Texaco to be the optimum for this engine with respect to swirl generation and combustion efficiency. The remaining parts of the piston assembly are standard L-141 items. In the selected chamber arrangement, the swirl increases substantially near the end of the compression stroke by transferring the rotating air charge into the small cup. As a result, a rather low intake swirl rate can be applied, which is desirable from a volumetric efficiency point of view.

The cylinder head and the fuel supply system used in the TCCS engine represent the two components which require the greatest modification from conventional spark ignition engine design practice. These modifications involve selection of the proper location and shape of the fuel injection nozzle, spark plug, and intake port design. Compared with a standard spark ignition engine, the cylinder head of the TCCS engine is rather simple, having a flat inner surface with valves located perpendicular to this surface. The fuel injection pump of the TCCS engine is a commercially available high-pressure diesel unit. Both the pump and the ignition distributor are driven by a specially designed drive mechanism.

Table 8-1. L-141 Engine Specifications, Standard and TCCS (Refs. 8-7, 8-8)

	Standard	TCCS
Configuration	In-line	
Cycle	4-stroke	
Cooling	Liquid	
Number of Cylinders	Four	
Total Displacement, in^3	141.5	
Bore, in.	3.875	
Stroke, in.	3.00	
Compression Ratio	7.5	10.0
Fuel	Combat Gas	Gas through Diesel No. 2
Induction System	Naturally Aspirated	Naturally Aspirated
Firing Order	1-3-4-2	1-3-4-2
Weight (dry with accessories), lb	328	375
Length, in.	26.9	27.5
Width, in.	21.4	24.5
Height, in.	23.0	23.3
Maximum Rated Power, Bhp	65(a)	64(b)
Rated Speed, rpm	4000	3200
Maximum Torque, lb-ft	110(a)	115(b)
Speed at Maximum Torque, rpm	1800	2400

(a) At 28.77 in Hg and 85°F, with air cleaner, generator, and automatic spark advance

(b) At 29.92 in Hg and 60°F, with air cleaner, generator, and automatic spark and injection advance

An emission-control system, consisting of EGR, oxidation catalysts, and intake-air throttling, was added later in the program at the expense of some loss in fuel economy. This system is further described in Section 8.2.7.2.2.

With the exception of these components, the design and construction of the TCCS engine is comparable to other reciprocating engines. The TCCS engine has been operated up to the maximum governed speed of 3800 rpm. Texaco feels that speeds up to 5000 rpm are feasible with TCCS by modifying the intake system and by utilizing a more favorable bore-to-stroke ratio.

8.2.2.6.3 Turbocharged L-141 TCCS Engine

Following the successful application of engine intake pressure charging in exploratory tests, Texaco has incorporated a commercially available turbocharger into some of its L-141 TCCS engines. Other changes to the engine were minor, with the most significant being a reduction of the compression ratio from 10:1 to 9.3:1, to reduce the structural loading of the lightweight L-141 engine and to minimize engine knock at full power (Ref. 8-8).

On the dynamometer, the engine has demonstrated good operating characteristics and is free from misfire and knock when operated on combat gasoline, CITE fuel, and No. 2 Diesel fuel. The effects of fuel variations on engine performance are further discussed in Section 8.2.7.1.

8.2.2.7 Ford Programmed Combustion Concept

8.2.2.7.1 General

Ford's Programmed Combustion (PROCO) concept represents a second-generation modification to the current reciprocating internal combustion engine. The principal design objectives of the first-generation Ford Combustion Process (FCP) engine were improved fuel economy compared with conventional spark ignition engines (without power loss), adequate cold start characteristics, a high degree of parts commonality with carbureted engines; smooth operating characteristics; and low emissions (Refs. 8-9 and 8-10). Research and development on this concept was initiated by Ford in 1956, and, starting in 1968, the design was applied to the L-141 military light-duty engine under TACOM sponsorship. Although the emissions of the original FCP engine were respectable, they fell far short of meeting long-range objectives.

Development of the second-generation PROCO engine was initiated in 1969 under joint TACOM and EPA sponsorship. In addition to meeting the objectives of the FCP engine, PROCO was designed to meet the 1976 statutory federal emission standards at low mileage with a minimum loss in fuel economy. These objectives have been achieved.

TACOM terminated its support of the L-141 PROCO program as of March 15, 1973 and is concentrating its efforts on development of the Texaco concept. However, work on PROCO and on carbureted modifications of the PROCO concept (Fast Burn) is being continued by Ford with company funding (Ref. 8-11). These efforts are aimed at the development of PROCO and Fast Burn engines for use in passenger cars. The design features and operational characteristics of these engines are outlined in the following sections.

8.2.2.7.2 PROCO Engine

The PROCO engine concept, as applied to the military L-141 light-duty engine, is illustrated in Figure 8-9. Engine specifications are listed in Table 8-2. In this engine, air enters through a special intake system which is designed to impart a moderate swirl around the cylinder bore axis in order to assist in attaining proper fuel-air mixing and good combustion. The air charge inducted into the cylinder is then compressed and transferred into the cup-shaped combustion chamber located concentrically in the piston. The top opening of the chamber has been optimized and represents 35 percent of the bore area, resulting in 65 percent squish action (Ref. 8-10).

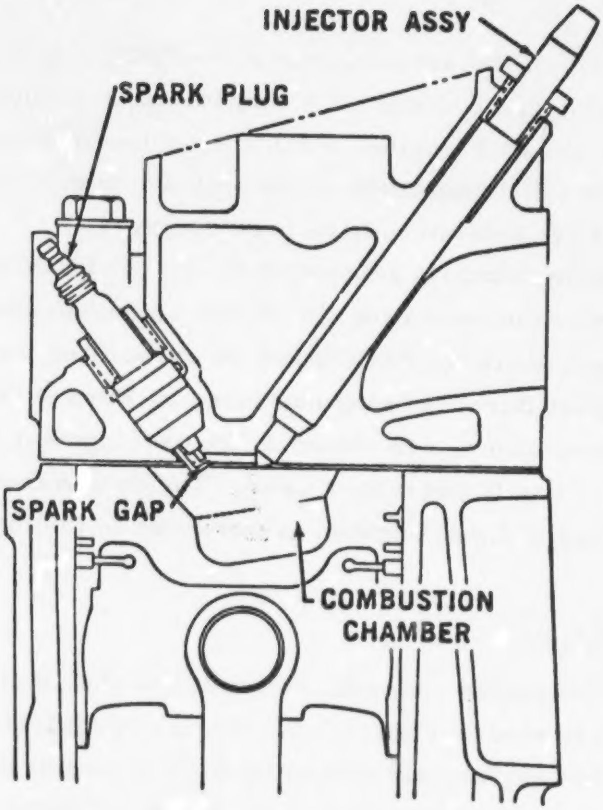

INJECTOR ASSY
SPARK PLUG
SPARK GAP
COMBUSTION CHAMBER

Figure 8-9. Ford L-141 PROCO Engine (Ref. 8-10)

Table 8-2. PROCO Specifications, L-141 Engine (Ref. 8-10)

Number of Cylinders	4
Cycle	Four-stroke
Displacement, CID	141.5
Compression Ratio	11.0
Fuel Injection System	Ford Experimental
Air-Fuel Ratio Control	Speed - Vacuum
Ignition System	Ford Transistorized
Spark Plugs	Experimental Short Electrode
EGR System	Fixed Orifice, water-cooled
Exhaust Manifold	Insulated, low inertia
Catalyst	Engelhard PTX-6 (noble metal)
Fuel	Lead Sterile; 91 RON

In addition, Ford has incorporated the PROCO concept into several of its 351-CID V8 engines. With the exception of a symmetrical piston cup and a dished cylinder head surface, the PROCO components used in this engine are equivalent to those employed in the L-141 engine (Ref. 8-10).

A compression ratio of 11.0 has been selected by Ford for all PROCO engines. Fuel is injected into each cylinder through an inclined nozzle configured to provide a low, penetrating, wide-angle (approximately 100 degrees), conical fuel spray. The spray mixes with the air and forms a rich mixture in its center surrounded by increasingly leaner mixtures and a layer of air adjacent to the cylinder wall to minimize quenching effects. A specially designed spark plug located just above the spray cone is used to ignite the mixture near the top-dead-center (TDC) position of the piston. From there, combustion proceeds very rapidly into the rich mixture cone and then into the leaner regions. As the piston descends, the swirling charge expands from the piston cup into the cylinder where the combustion process is then carried to completion.

The NO_x emissions of the engine are further reduced by means of exhaust gas recirculation (EGR), which is inducted into the air intake after passing through a water-cooled heat exchanger. The HC and CO emissions are controlled by a system consisting of a low-inertia, insulated manifold; an oxidation catalyst, and air throttling.

The HC, CO, and NO_x emissions from the PROCO engine are inherently low due to the stratified charge combustion process. HC is minimized internally through the use of late injection in conjunction with a soft fuel spray used to prevent fuel deposition on the cylinder wall. CO is inherently low because of the availability of excess air at all operating conditions except at full power. NO_x is controlled by initiating combustion in the fuel-rich zone, which results in moderate flame temperature and oxygen-free, post-flame gases. In addition, the engine is operated with high EGR rates and late combustion to further reduce the combustion temperature and the NO_x formation rates. PROCO engine and vehicle emission performance and test data are discussed in Sections 8.2.7.2.3 and 8.2.7.3.2.

8.2.2.7.3 Fast Burn Concept

The Ford Fast Burn engine concept is a modification of the PROCO engine in which the fuel injection system has been replaced by a carburetor. To date, two configurations, Fast Burn Phase I and Fast Burn Phase II, have been tested by Ford (Ref. 8-11).

Fast Burn Phase I consists of a bowl-in-piston combustion chamber similar to the PROCO design. The engine has a compression ratio of 11.0 and utilizes high air swirl rates in conjunction with improved carburetion, EGR, and modified spark advance. This concept was tested in a modified 351-CID engine installed in a 1972 Torino automobile. The emission and performance data from these tests are presented in Sections 8.2.7.2.3 and 8.2.7.3.2.

The design features of the Fast Burn Phase II concept are illustrated in Figure 8-10. As indicated, the engine incorporates a conventional combustion chamber and two spark plugs per cylinder and utilizes EGR, low-intake air swirl, and a normal production/compression ratio. Initial concept evaluation tests have been conducted on a 400-CID V-8 engine installed in a 1973 Torino vehicle.

8.2.3 Fuel Injection and Ignition Systems

Fuel injection in both the TCCS and PROCO engines is aimed at creating a localized fuel-rich zone in the vicinity of the piston cup. In the PROCO engine, injection occurs early in the compression stroke so that rich zone mixing can occur before ignition. To retain combustion stability over a suitably large speed and load range, the injection timing advance is varied over a wide range as a function of speed and load. Commercially available diesel injection systems do not have provisions for such wide timing advance schedules and they also inject much faster than optimum under the low pressure injection conditions of the PROCO. Therefore, in 1961 Ford initiated development of a specialized injection pump and nozzles tailored to its combustion process. The new low-pressure (250 to 350 psi) fuel injection

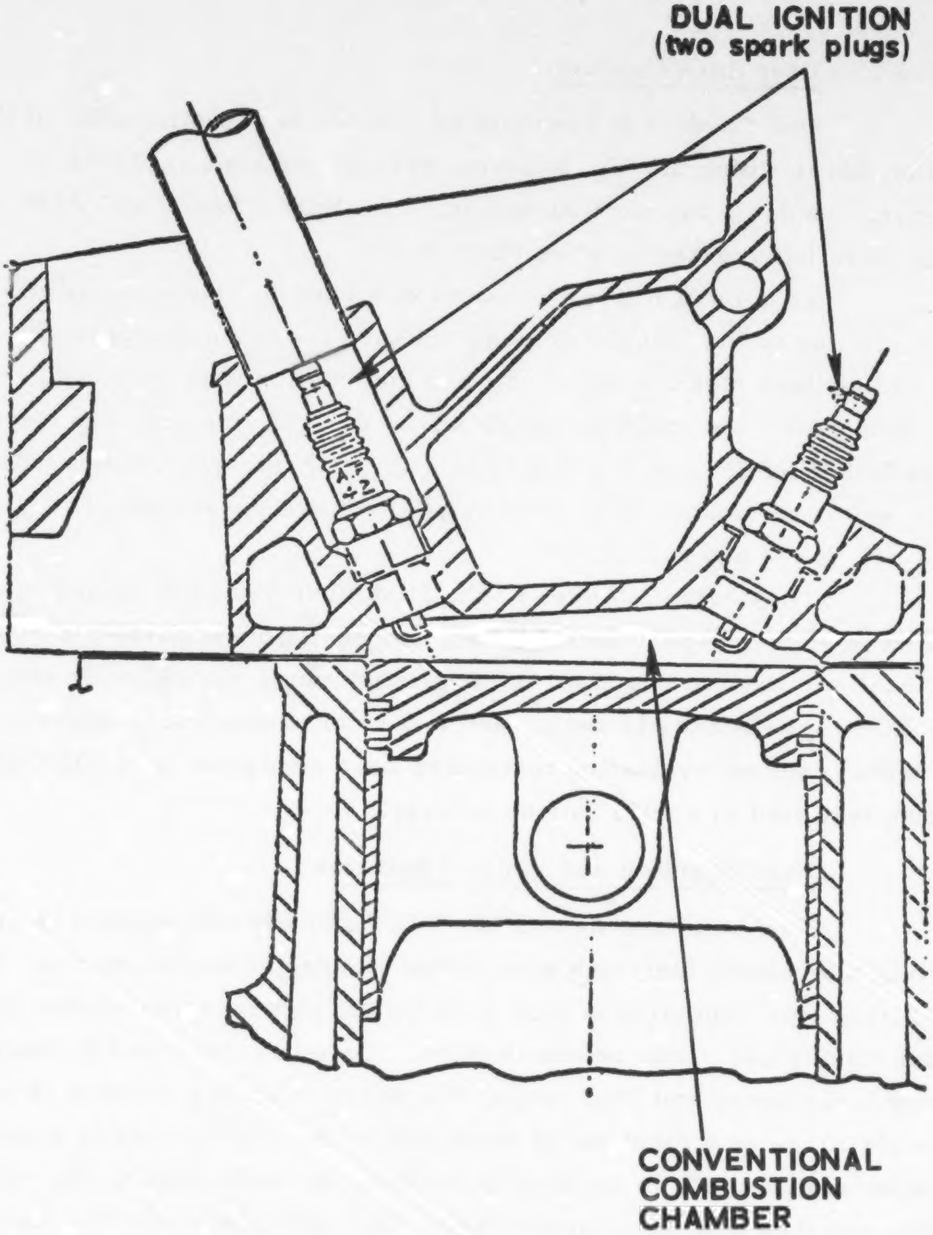

DUAL IGNITION
(two spark plugs)
CONVENTIONAL
COMBUSTION
CHAMBER

Figure 8-10. Ford Fast Burn Phase II Stratified Charge Engine (Ref. 8-11)

system consists of individual plungers for each cylinder, a metering sleeve for flow control, and a poppet valve type injector designed to vibrate for improved atomization.

In the TCCS engine, the fuel is injected at high pressures provided by a commercially available fuel injection system (Ref. 8-12).

The ignition systems for the two engines also show significant differences. In PROCO, ignition starts late (essentially at top dead center), and lasts for only about 3 degrees of crank, permitting the use of a more or less conventional ignition system. Because fuel injection and spark timing schedules must be coordinated accurately, the PROCO has a unitized injection pump/distributor. While this makes for a seemingly complex mechanical design, the control elements are sealed and protected, and the unit operates with low fuel pressure and low mechanical loadings.

In the earlier FCP engines, Ford utilized long spark plugs with electrodes extending to the center of the fuel spray. However, with air throttling and EGR, shorter electrodes proved to be superior in PROCO with respect to emissions, fuel consumption, and plug life.

In the Texaco process, sparking is initiated almost simultaneously with fuel injection and continues throughout the duration of the injection cycle. Positive ignition is therefore assured, but a specialized ignition system is required. The Texaco ignition system features a fast rise time, a 75,000-volt spark, and multiple sparking (as much as 100 restrikes over a 15 to 20 degree crank angle).

8.2.4 Fuel Requirements

Texaco's naturally aspirated and turbocharged L-141 TCCS engines have been operated repeatedly on various fuels ranging from high octane gasoline to high cetane diesel fuel without incurring engine misfire or knock. Also, the specific fuel consumption and exhaust emissions have remained essentially unchanged when fuels are switched (Ref. 8-13).

Conversely, the PROCO engine is susceptible to combustion knock. However, test data have indicated that the octane sensitivity of PROCO is less than in conventional spark ignition engines, and 91 RON gasoline is acceptable. The relatively low octane requirement of PROCO engines is attributed to the very lean air-fuel mixture of the end gas and the high EGR rates utilized in this engine.

Since oxidation catalysts are needed for HC and CO control in these stratified charge engines, lead-free fuel is required to prevent catalyst poisoning.

8.2.5 Power Control Requirements

The power control requirements and methods utilized in stratified charge engines are comparable to those in conventional spark ignition engines.

In the PROCO engine, the injection function incorporates a compound control system consisting of a speed/density, air fuel ratio control at idle and part load; a mechanical enrichment system for maximum power; and a fuel shutoff valve for deceleration. With increasing power demand, the air throttle is gradually opened to the wide-open position during the first two-thirds of the accelerator pedal movement while the EGR valve is kept fully open. During the last third of the pedal movement, the EGR flow rate is gradually reduced to zero for the full power setting (Ref. 8-10).

In the most recent TCCS engine installations, power is controlled by the accelerator pedal, which is connected to the injection pump and the automatic transmission. Conversely, the earlier configurations employed intake throttling at idle and during deceleration, as well as modulated EGR.

8.2.6 Starting Characteristics

In general, open-chamber stratified charge engines exhibit good starting characteristics.

Controlled cold-starting tests conducted on the Texaco TCCS engine at temperatures as low as -20°F indicate excellent starting response on gasoline. With CITE fuel, successful starts are usually achieved within

10 seconds of engine cranking. However, starting problems were occasionally encountered with diesel fuel (Ref. 8-7).

Ford's stratified charge engines have always exhibited excellent starting characteristics. Also, driveaway following a cold start has never been a problem on these engines. However, after prolonged periods of standing, the engine may idle rough for several seconds, until all the air is purged from the fuel injection system (Ref. 8-9).

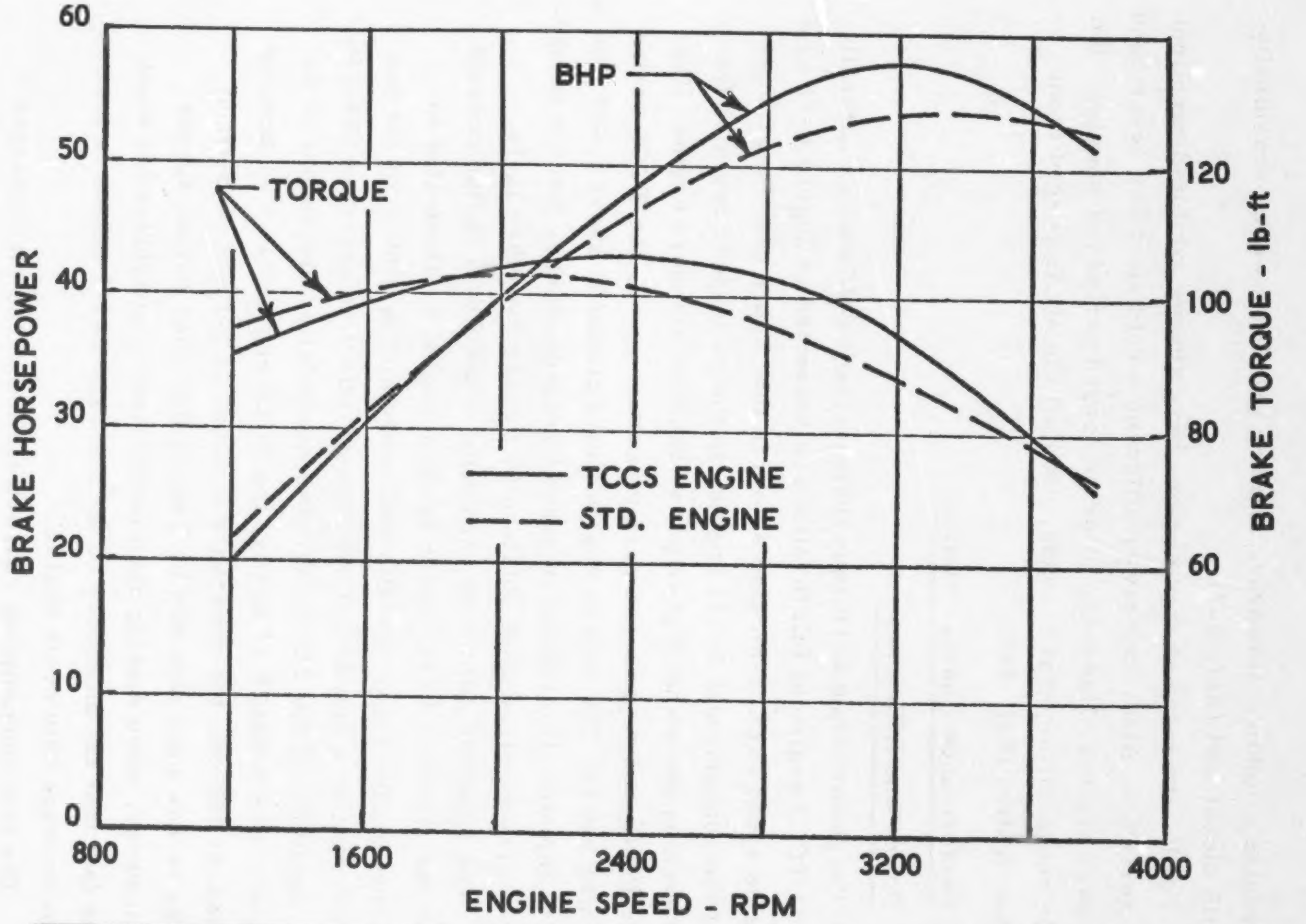
8.2.7 Performance Characteristics

8.2.7.1 Power and Efficiency

The power output and torque characteristics of Texaco's naturally aspirated L-141 TCCS engine at full throttle are presented in Figure 8-11 as a function of engine speed (Ref. 8-8); also shown in this figure are the performance curves of the conventional L-141 engine. In the mid-speed regime, the TCCS engine develops somewhat higher power than the standard engine. Conversely, in the low-speed regime and near maximum speed, the TCCS engine has slightly lower power. The loss in engine power at speeds above 3,200 rpm is due primarily to injection limitations imposed by engine smoke. Similar smoke-limited power characteristics were obtained by Texaco for other fuels.

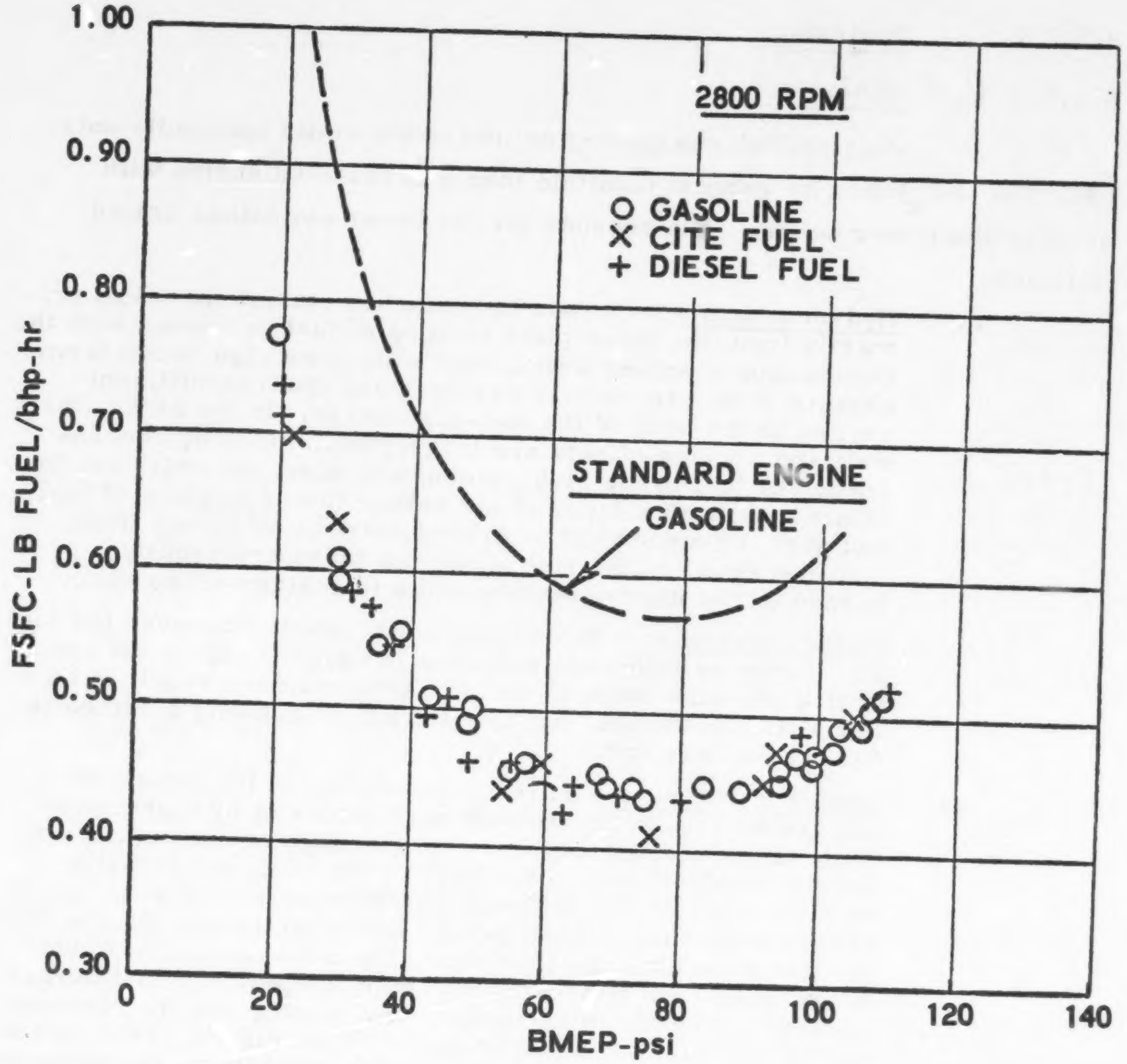
The superior efficiency capability of the TCCS engine concept with respect to the standard L-141 spark ignition engine is illustrated in Figure 8-12, showing the brake specific fuel consumption (BSFC) of the two engines at 2,800 rpm as a function of brake mean effective pressure (BMEP). As indicated, the BSFC of the TCCS engine is essentially independent of the type of fuel used. For a BMEP of 30 psi, the TCCS engine has a 27 percent lower BSFC and, at 100 psi, the difference is about 24 percent in favor of TCCS. Similar trends were obtained by Texaco for other engine speeds (Ref. 8-8). However, when making such comparisons, consideration must be given to the fact that the efficiency of the standard L-141 engine is lower than that of the average Otto cycle engine.

The fuel consumption characteristics of the turbocharged TCCS engine and of Ford's PROCO engine are comparable to the naturally aspirated TCCS engine.



MAP-29.0" HG IAT-85° F DYN. EXHAUST AIR CLEANER
GENERATOR AUTOMATIC SPARK AND INJECTION ADVANCE

Figure 8-11. Full-Load Brake Performance, Texaco Naturally Aspirated L-141 TCCS Engine, Gasoline (Ref. 8-8)



29.0" HG, AMBIENT PRESS. 85°F, AMBIENT TEMP. DYN. EXHAUST
AIR CLEANER
GENERATOR
AUTOMATIC SPARK & INJECTION ADVANCE

Figure 8-12. Multifuel Brake Performance, Texaco Naturally Aspirated L-141 TCCS Engine (Ref. 8-8)

8.2.7.2 Emissions

8.2.7.2.1 General

A stratified-charge engine inherently emits less pollutants (HC, CO, NO_x) into its exhaust manifold than a carbureted engine with comparable power output. The reasons for the lower emissions are as follows:

a. Hydrocarbons - Unburned hydrocarbon emissions result primarily from the incomplete burning of fuel in contact with the combustion chamber walls. HC emissions also result from capture of fuel in various crevices and from insufficient oxygen in the body of the fuel-air charge. In the SCE,* the wall and crevice effects are largely eliminated by concentration of fuel in the rich-combustion zone; the walls are in effect lined with a layer of air rather than a mixture of fuel and air. Unburned hydrocarbons exist in relatively high concentrations in the rich zone, but these are readily burned by the excess oxygen in the lean stage of the burn.

b. Carbon Monoxide - Emissions of CO result whenever the fuel-air charge is uniformly deficient in oxygen. As in the case of HC, the rich stage of the SCE process does result in high initial CO formation, but this is very effectively oxidized to CO_2 at the lean end.

c. Oxides of Nitrogen - Three factors play an important and integrated role in the formation of oxides of nitrogen: peak cycle temperature, exposure time at high temperature, and oxygen availability. In the case of the SCE, the initially rich combustion limits the peak temperature and assures an oxygen-free environment behind the flame front. By the time the flame reaches the lean zone, the temperature is reduced, so the residence time with both high temperature and oxygen present is greatly minimized. As a result, the NO_x formed during combustion is rather low. These effects are depicted in Figure 8-13. In the early stages of combustion, the three pollutants are formed in concentrations shown on the left of the figure. By the end of the combustion cycle, the concentrations correspond to the right of the figure, with the transition analogous to "tunneling" across the figure. Particularly in the case of NO_x, the conditions corresponding to near-stoichiometric combustion are not permitted to exist long enough for NO_x to build up to equilibrium levels.

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*Stratified-Charge Engine.

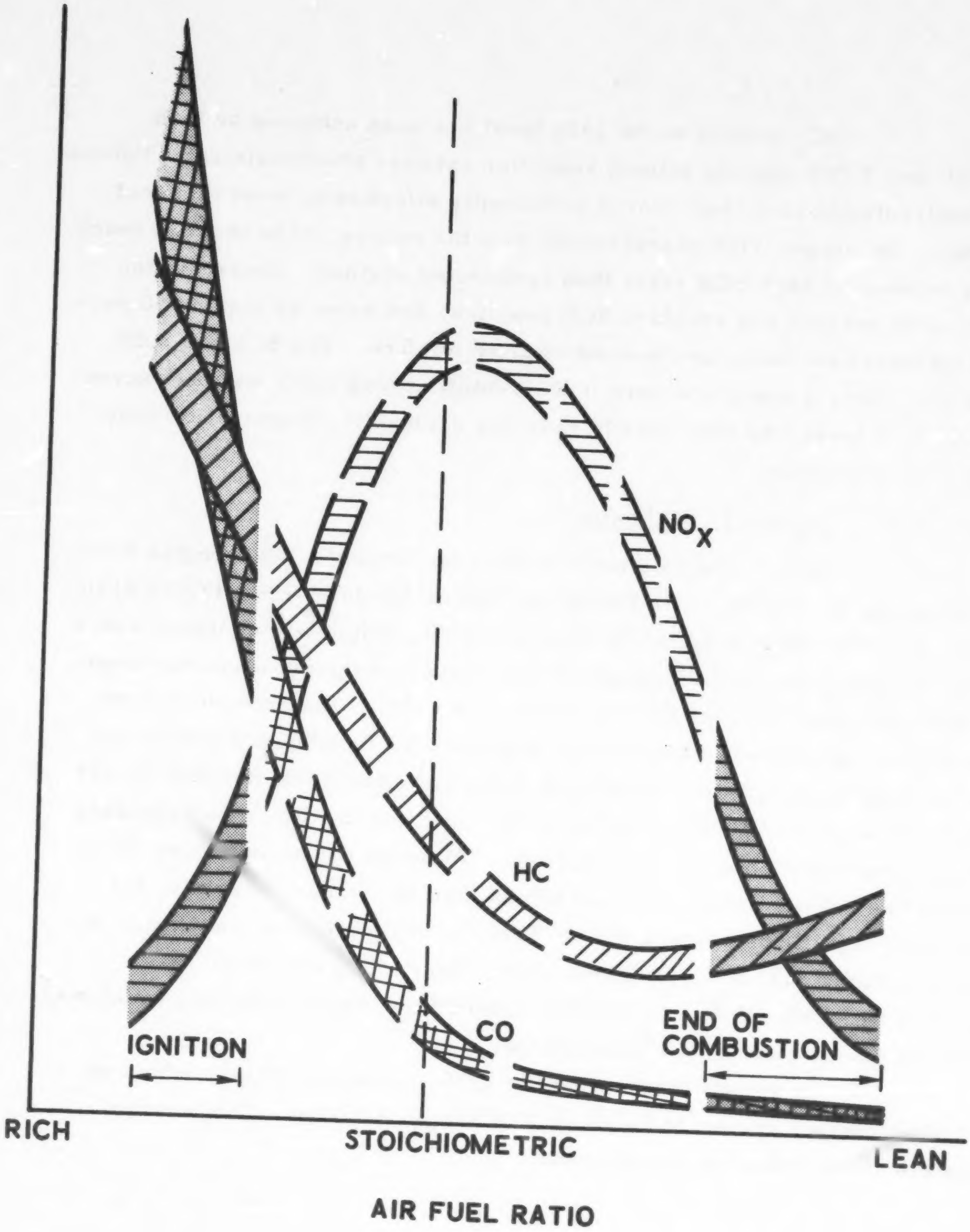


Figure 8-13. Effect of Stratified Charge Combustion on Emissions

NO_x control to the 1976 level has been achieved by both PROCO and TCCS engines without reduction catalyst aftertreatment. Exhaust gas recirculation alone has proven sufficiently effective to meet this goal. Further, the oxygen-rich characteristics of the exhaust make the SCE much more tolerant of high EGR rates than carbureted engines. Recirculation rates of 25 percent are standard SCE practice, and rates as high as 50 percent (at idle) have been used without causing misfire. The SCE has sufficient NO_x control margin to permit EGR cutoff at high loads without exceeding the 1976 level, thereby greatly reducing driveability degradation under peak-load conditions.

8.2.7.2.2 Texaco TCCS Engine

Initial vehicle emission tests on Texaco's TCCS engine were conducted by the National Air Pollution Control Administration (NAPCA) in 1968. In 1970, NAPCA tested an M-151 vehicle, which was equipped with a turbocharged L-141 TCCS engine and was driven on the dynamometer over the then proposed Federal Driving Cycle (LA 4-S3). The emissions from this engine, measured in accordance with the CVS technique (3,000-pound inertia weight), are listed along with the emissions of the standard M-151 vehicle in Table 8-3. Both vehicles were tuned for maximum performance with no consideration given to emissions. Although the uncontrolled TCCS vehicle has substantially lower emissions than the standard vehicle, the levels are significantly above the 1975-76 Federal emission standards. As shown in Table 8-3, some reduction in the HC and CO emissions was achieved by means of a commercially available oxidation catalyst, which was installed downstream of the turbocharger.

In an attempt to meet the 1976 standards, Texaco added an emission control system to its basic TCCS engine design. This system consists of the following components:

a. A cooled, dual-rate EGR system for NO_x control, incorporating a particulate trap, an EGR cooler, and an EGR mixer. At high loads, the EGR flow rate is about 15 percent of the air charge and at low loads the rate is somewhat higher.

b. A closely coupled platinum catalyst incorporating a turbulence device.

c. A second platinum catalyst and a copper chromite catalyst installed downstream of the closely coupled catalyst.

d. Intake air throttling at idle and low loads in order to increase the exhaust temperature and improve the catalyst conversion efficiency at the expense of some loss in fuel economy.

Table 8-3. Exhaust Emissions of a Turbocharged M-151 TCCS Vehicle (Ref. 8-8)

Emissions,* gm/mi	M-151 TCCS no catalyst	M-151 TCCS with catalyst	Standard M-151
HC	3.85 - 4.58	1.74 - 1.97	6.4
CO	9.08 - 9.62	2.29 - 2.69	76.2
NO_x	1.74 - 1.92	1.81 - 2.23	3.4

*LA 4-3S Test Cycle
3,000-lb Inertia Weight
Cold Start
Gasoline Fuel

The M-151 vehicle incorporating this baseline emission control system was tested by the EPA and by Texaco in 1972, and the test results are presented in Table 8-4. As indicated, vehicle emissions are below the original 1976 statutory Federal emission standards. However, it should be noted that the vehicle was not able to either negotiate all the acceleration modes in the cycle or achieve the maximum cycle speed of 57 mph.

Table 8-4. Exhaust Emissions, Controlled Naturally Aspirated M-151 TCCS Vehicle Equipped with Exhaust Gas Recirculation System and Oxidation Catalysts

Laboratory	Mass Emissions - gm/mi HC	 CO	 NO_x
Texaco	0.25	1.17	0.33
Texaco	0.28	0.62	0.32
Texaco	0.40	0.26	0.29
EPA	0.40	0.26	0.30
EPA	0.33	0.15	0.31
EPA	0.37	0.30	0.31
1975 CVS C/H 2,750-lb Inertia Lead-free Gasoline			Refs. 8-8, 8-14

Upon completion of a 50,000-mile durability run, the vehicle was retested by the EPA, and the emissions were lower than the 1976 statutory emission standards. However, major maintenance work was required on the engine during the durability test, including several ignition system modifications, replacement of four catalysts, cleaning of the EGR system, and major valve train maintenance (Ref. 8-15).

The effect of emission control system modifications on the emissions and fuel economy of the baseline TCCS system incorporated into an M-151 vehicle is illustrated in Table 8-5. The modifications, which were implemented in 1972, include fuel injection and ignition retardation, various levels of modulated EGR, and utilization of oxidation catalysts in the engine exhaust. As indicated in the table, the CO and NO_x emissions of both the uncontrolled naturally aspirated, and turbocharged engines are low, compared with uncontrolled conventional gasoline engines. Conversely, with respect to HC,

Table 8-5. Effect of Emission Control System Modifications on M-151 TCCS Vehicle Emissions and Fuel Economy, 1975 FTP

Test Lab	Case No.	Engine Configuration	Turbo-charged	Emissions* - gm/mi			Fuel Economy, mi/gal by wt.	Max. Rear Wheel hp at 50 mph	References
				HC	CO	NO_x			
Texaco	1	Max. economy setting, no catalyst	Yes	3.13	7.00	1.46	24.3	40.0	8-13
Texaco	2	Max. economy setting, no catalyst	No	4.24	7.28	1.43	-	-	8-13
Texaco	3	8-deg combustion retard only, no catalyst	Yes	3.24	6.43	1.29	22.4	39.3	8-13
Texaco	4	8-deg combustion retard, low EGR rate, no catalyst	Yes	3.60	6.69	0.84	20.5	38.0	8-13
Texaco	5	8-deg combustion retard, medium EGR rate, two platinum catalysts	Yes	0.33	1.05	0.61	19.7	36.0	8-13
Texaco	6	13-deg combustion retard, high EGR rate, two platinum catalysts	Yes	0.35	1.41	0.35	16.2	28.7	8-13
Texaco	7	5-deg combustion retard, no EGR, two platinum catalysts, alternate vehicle	Yes	0.30	1.07	1.40	20.9	40.0	8-13
Texaco	8	Baseline configuration**	No	0.36	0.61	0.31	16.2	-	8-8
Texaco	9	Baseline configuration with reduced EGR	No	0.48	0.57	0.45	17.6	-	8-8
EPA	10	Baseline configuration**	No	0.37	0.24	0.31	15.8	-	8-8
EPA	11	Baseline configuration with reduced EGR	No	0.50	0.14	0.70	21.9	-	8-8

*Fuel - Gasoline 91 RON, lead-free, + 2% oil.

**Exhaust Gas Recirculation (EGR) plus two platinum catalysts the TCCS engines are comparable to conventional spark ignition engines. Some reduction in HC and CO can be achieved by retarded spark timing at the expense of lower fuel economy. As in conventional engines, EGR is quite effective in reducing NO_x, and the use of oxidation catalysts results in a substantial reduction of HC and CO. Similar results were obtained with JP-4 and diesel fuel (Ref. 8-8).

Emission data from a TCCS-powered Plymouth Cricket vehicle tested in 1973 by Texaco, in accordance with the 1975 Federal Test Procedure (FTP), are listed in Table 8-6. The emission control system incorporated in the engine consisted of modulated cooled EGR, two oxidation catalysts, and air throttling at low loads. As indicated, the average emissions obtained in these tests are below the statutory 1976 standards (Tests 30, 32, and 35). While CO is attractively low, Chrysler feels that HC and NO_x are not low enough to provide an acceptable allowance for system deterioration due to mileage accumulation and production tolerances (Ref. 8-16). According to Chrysler, additional development is necessary to simplify the system design and to improve the EGR control.

Emission and fuel economy data obtained on this vehicle by Texaco are also listed in Table 8-6. The data demonstrate that injection and ignition retardation are quite effective in reducing NO_x emissions at the expense of some loss in fuel economy. When adjusted for best economy, the fuel consumption of the TCCS-powered vehicle is about 20 percent lower than the average 1973 Cricket certification vehicle.

8.2.7.2.3 Ford PROCO and Fast Burn Engines

Results from M-151 PROCO vehicle tests conducted in 1971 by Ford (Ref. 8-10) and by the EPA (Ref. 8-17) are listed in Table 8-7. In these tests, the EGR cutoff and fuel enrichment at high power levels were deactivated to minimize NO_x. As a result, the vehicle had insufficient power to negotiate the two highest acceleration modes of the Federal Driving Cycle. Ford estimates that a 20 to 25 percent larger engine would develop sufficient power to follow the cycle with EGR.

Table 8-6. Exhaust Emissions from a Chrysler Cricket TCCS Vehicle, 2,500-lb Inertia Weight (Ref. 8-16)

Test No.	Timing - Injection and Ignition	Cata-lysts	EGR	EBP*	Air Throt-tled	1975 EPA gm/mi			Fuel Economy mpg
						HC	CO	NO_x	
Avg. 3 (30, 32 & 35)	Retarded for low emissions	Yes	Yes	Yes	Yes	0.36	1.15	0.38	20.1
44	Retarded for low emissions	Yes	Yes	No	No	0.59	0.64	0.54	22.6
47	Retarded for low emissions	Yes	No	Yes	No	0.61	0.85	0.99	22.5
50	Retarded for low emissions	Yes	No	No	No	0.73	1.18	1.20	23.8
53	For best economy	Yes	No	No	No	1.07	0.84	1.89	25.3

*EBP = Exhaust Back Pressure Orifice

Table 8-7. Emissions and Fuel Economy, M-151 PROCO Vehicles, 1975 CVS Test Procedure (References 8-10 and 8-17)

Vehicle	Emissions,(3) gm/mi			Fuel Economy,(1)(4) mi/gal	Number of Tests (averaged)	Test Facility
	HC	CO	NO_x			
PROCO, no catalyst	2.60	13.45	0.32	21.7	1	Ford
PROCO, with catalyst	0.35	1.01	0.35	21.3(2)	4	Ford
PROCO, with catalyst	0.37	0.93	0.33	Not measured	14	EPA
1972 CVS Test Procedure						
PROCO, best fuel economy, no catalyst	4.96	7.75	3.85	23.8	2	Ford
PROCO, no catalyst	3.10	13.75	0.33	21.2	1	Ford
PROCO, with catalyst	0.54	1.18	0.37	19.6	4	Ford

CVS = constant value sampling.

(1) Computed from the mass emission data (miles per gallon).

(2) 20.4 miles per gallon, measured with a burette.

(3) Baseline (carbureted) vehicle emissions are 4.55 gm/mi HC, 41.6 gm/mi CO, 4.4 gm/mi NO_x.

(4) Baseline (carbureted) vehicle fuel economy 17.2 mi/gal.

As indicated in Table 8-7, the low-mileage emissions achieved with the derated M-151 PROCO vehicle with catalyst are below the 1976 statutory federal emission standards. Ford emphasizes, however, that this was accomplished with carefully controlled engine adjustments derived from a long series of optimization tests (Ref. 8-10). Owing to the high EGR rates employed, the engine operated close to its misfire limit. Without catalyst but with EGR, the HC emissions of the PROCO engine are about 40 percent lower than for the carbureted configuration, and CO is about 70 percent lower. At the best economy setting, PROCO gives about 10 percent lower HC and NO_x and about 85 percent less CO.

In 1973, Ford conducted AMA durability tests of a 1972 Montego automobile equipped with a 351-CID PROCO engine and two Engelhard PTX oxidation catalysts. As shown in Table 8-8, the low mileage

Table 8-8. 351-CID PROCO Durability -- Vehicle: 1972 Montego, 110T722, 1975 Test Procedure (Ref. 8-11)

	Catalyst Mileage	Emissions, gm/mi			Fuel Economy[1] mpg
		HC	CO	NO_x	
At the start	10	0.16	0.25	0.32	13.9
At 26,000 miles, after repair and readjustment	26,195	0.79	1.16	0.37	12.75[2]
At 26,000 miles, after application of part-load back-pressure system and 240° thermostat	26,400	0.47	1.11	0.35	13.2

[1] Based on carbon balance

[2] Tank mileage for 26,000 miles: 12.7 mpg

emissions are below the statutory 1976 standards. However, after accumulation of 26,000 miles, the HC and CO emissions had increased significantly, indicating excessive catalyst performance degradation. Subsequent installation of a 240°F thermostat, combined with higher backpressure operation of the engine at part load, resulted in a substantial reduction in HC (Ref. 8-11).

The effect of EGR on the emissions and fuel economy of this vehicle, tested without catalysts, is illustrated in Table 8-9. A reduction of the EGR flow rate from 25 percent to 15 percent resulted in substantially lower HC emissions at the expense of higher NO_x. Apparently, the variations in EGR performed by Ford had little effect on fuel economy.

Table 8-10 presents emission test data obtained by Ford from a 351-CID V-8 Fast Burn Phase I engine installed in a 1972 Torino vehicle (Ref. 8-11). With oxidation catalyst, the vehicle meets the 1976 statutory emission standards for CO and NO_x at low mileage and exceeds the HC standard slightly. However, after 25,000 miles, the HC and CO emissions

Table 8-9. 351-CID PROCO Emissions and Fuel Economy -- Vehicle: 1972 Montego, 110T722, 1975 Test Procedure (Ref. 8-11)

NO_x Design Level (gr/mi)	EGR %	Emissions, gm/mi			Fuel Economy[1] mpg	Remarks
		HC	CO	NO_x		
0.4	25	3.54	28.2	.34	13.8	no catalyst
1.5	15	1.34	25.0	.96	13.2	no catalyst

(1)Based on carbon balance

Table 8-10. Vehicle Emissions and Fuel Economy, 351-CID Fast Burn Phase I Engine in a 1972 Torino, 1975 Test Procedure, Average of Two Tests (Ref. 8-11)

Mileage	Emissions, gm/mi			Fuel Economy mpg	Remarks
	HC	CO	NO_x		
Low	6.9	78.3	.37	11.9	w/o catalyst
Low	.44	1.31	.34	11.9	w/HC-CO catalyst
25,154	2.52	5.78	.37	11.4	w/HC-CO catalyst

are substantially higher than the 1976 standards. The rapid deterioration of the catalyst is attributed to the high HC and CO levels of the raw engine exhaust, resulting in high thermal loading of the catalyst.

Fast Burn Phase II vehicle emission data are presented in Table 8-11 for operation without catalyst and secondary air, but with EGR. Under these conditions, NO_x is rather low but HC and CO are excessive. The data show again the previously noted interrelationship between NO_x, HC, CO, and fuel economy.

Table 8-11. Vehicle Emissions and Fuel Economy, 400-CID Fast Burn Phase II Engine in a 1973 Torino, 1975 Test Procedure (Ref. 8-11)

NO_x Design Level	Emissions, gm/mi			Fuel Economy mpg	Remarks
	HC	CO	NO_x		
0.40	3.1	30	0.66	8.6	w/o catalyst or secondary air
1.5	2.0	7.1	1.5	11.2	w/o catalyst or secondary air

Ford has conducted a considerable amount of single-cylinder PROCO engine test work to determine the effects of various operating parameters on emissions and fuel consumption (Ref. 8-10). The parameters investigated include injection and ignition timing, overall air fuel ratio, and EGR. As expected, retarded ignition and/or injection timing and EGR reduce NO_x at the expense of higher HC, CO, and fuel consumption.

8.2.7.3 Fuel Economy

8.2.7.3.1 Texaco TCCS Engine

As discussed in Section 8.2.2.1, an emission controlled M-151 TCCS vehicle was tested by the EPA in accordance with the 1975 FTP and after the vehicle had been subjected to a 50,000-mile durability run (Ref 8-15). The emission control system utilized on this vehicle consisted of EGR, three catalysts, and intake throttling. It also required considerable maintenance work during the test period. The fuel economy, determined by the EPA by means of the carbon balance method from six consecutive tests, varied between 14.4 mpg and 17.1 mpg with the average at 15.5 mpg.

Somewhat better fuel economy data have been reported by Texaco for the standard emission control system and for modified configurations (Ref. 8-8). These data, listed in Table 8-5, show that substantial fuel economy and power losses are associated with the achievement of extremely low NO_x emission levels. For example, a fuel economy loss of 33 percent and a power loss of 28 percent are incurred in achieving the 1976 Federal emission standards. Catalytic treatment of the exhaust gases combined with moderately retarded combustion results in a 5 to 10 percent fuel economy degradation (Table 8-5, case 11). As illustrated in Table 8-6, similar results were obtained by Texaco for a TCCS-powered Plymouth Cricket automobile equipped with EGR and oxidation catalysts and tested with a 2,500-pound inertia weight. The best fuel economy obtained with this vehicle without EGR was 25.3 mpg. This compares with 20.6 mpg for the average 1973 model year Cricket certification vehicle tested at 2,250-pound inertia weight (Ref. 8-18).

Level road fuel economy data for the naturally aspirated L-141 TCCS engine, installed in an M-151 light-duty vehicle, are presented in Figure 8-14 (Ref. 8-8). The fuel economy of the TCCS-powered vehicle is considerably better than the fuel economy of the vehicle with the standard engine, especially at low vehicle speeds. For example, at 30 miles per hour, TCCS has a 30 percent better fuel economy, and at 55 mph, the difference is still 20 percent. The variations shown in Figure 8-13 for the different fuels are directly related to the fuel heating value per gallon and confirm the previously noted trends of specific fuel consumption (Figure 8-12).

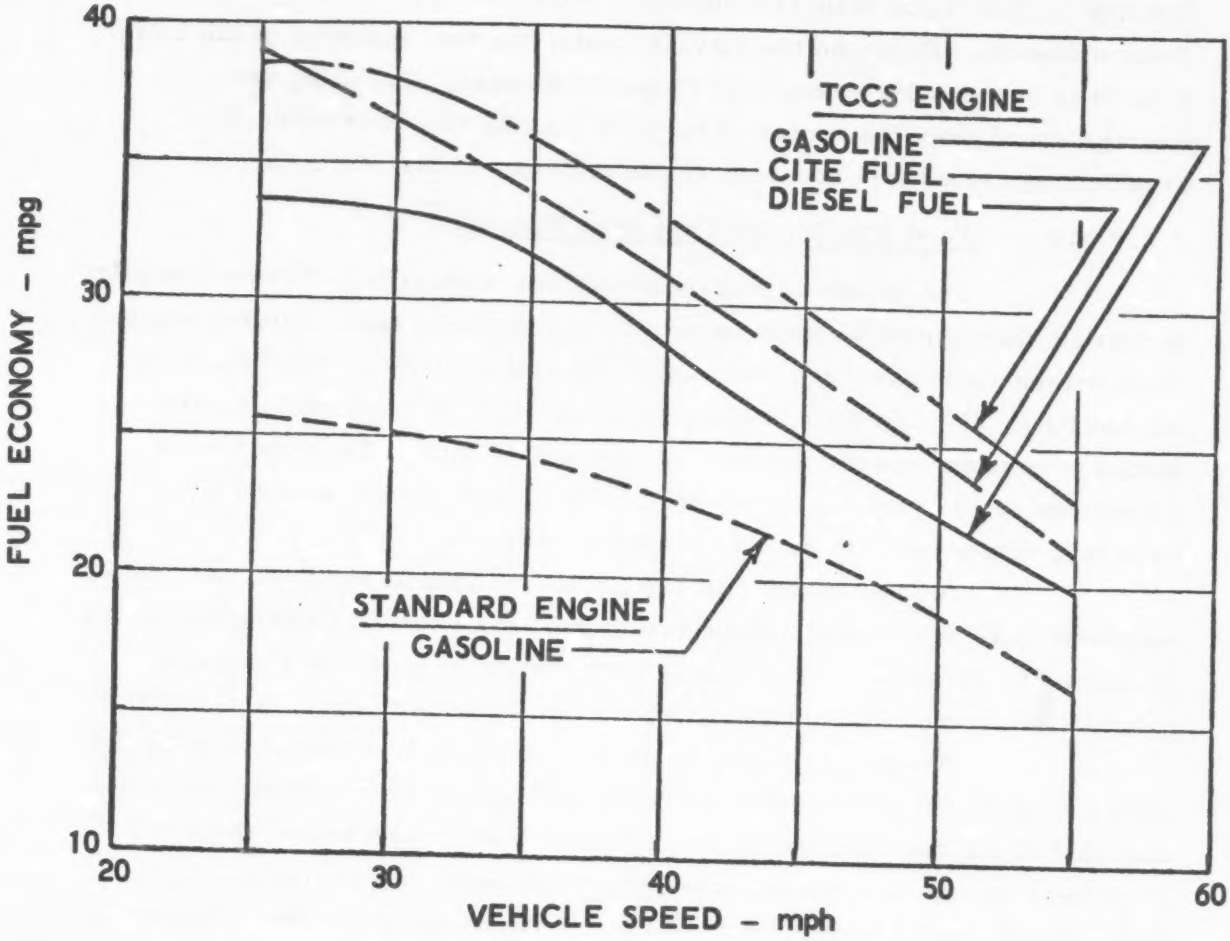


Figure 8-14. Level-Road Fuel Economy, M-151 Light-Duty Vehicle, Texaco Naturally Aspirated, L-141 TCCS Engine (Ref. 8-8)

Except for vehicle speeds below about 30 mph, the level-road fuel economy of the turbocharged M-151 TCCS vehicle is about 5 to 25 percent better than that of the naturally aspirated TCCS engine, with the higher improvement realized at vehicle speeds of 50 mph and above (Ref. 8-8).

Fuel economy tests over a closed-road course conducted by TACOM on TCCS and standard-engine-powered M-151 vehicles support the Texaco results. Based on the TACOM tests, the fuel economy of the TCCS vehicle is between 39 percent and 73 percent better, depending upon the average speed over the course (Ref. 8-8). Again this illustrates the excellent part-load fuel economy characteristics of the TCCS engine.

8.2.7.3.2 Ford PROCO and Fast Burn Engines

The improved fuel-economy characteristics of open-chamber, stratified charge engines with respect to conventional spark ignition engines were demonstrated by Ford early in its development program (Ref. 8-9). At that time, 430-CID FCP engines were installed in three vehicles and tested at constant speeds between 30 mph and 70 mph. The data show a 28 percent to 34 percent improvement in the fuel economy of the FCP vehicles, compared with similar standard vehicles.

Fuel economy data for the M-151 vehicle equipped with conventional and PROCO engines and tested over the Federal Driving Cycle are presented in Table 8-7. The values were computed from the measured exhaust-gas composition utilizing the carbon-balance method (Ref. 8-10).

The fuel economy of the PROCO vehicle adjusted to meet the 1976 statutory emission standards at low mileage is about 24 percent better than that of the conventional engine powered M-151 vehicle and about 12 percent better than the average 1973 model year vehicle tested at 2,750-pound inertia weight (Ref. 8-19). When adjusted for best efficiency, the fuel economy of PROCO is approximately 36 percent better than the average 1973 certification car tested at 2,750-pound inertia weight. However, this improvement results in a substantial increase of the HC, CO, and NO_x emissions.

Fuel economy data obtained by Ford for its PROCO engine equipped 1972 Montego vehicle are listed in Tables 8-8 and 8-9. The computed fuel economy of this vehicle over the Federal Driving Cycle is about 24 percent better than that of the average 1972 certification vehicle in the 4,500 pound weight class.

The fuel economy of the Fast Burn Phase I, 351-CID engine -- installed in a 1972 Torino automobile and tested according to the 1975 Federal Test Procedure -- is listed in Table 8-10. As indicated, the vehicle meets 1976 emission standards at low mileage with a definite improvement in fuel economy over a standard Torino. After 25,000 miles, the fuel economy of the vehicle is essentially unchanged, although the HC and CO emissions have increased considerably (Ref. 8-11).

As shown in Table 8-11, the fuel economy of Ford's Fast Burn Phase II vehicle adjusted to low NO_x levels is not very impressive and is equivalent to conventional 1973 Torino cars. However, the fuel economy improves substantially by adjusting the NO_x emissions to the 1.5-g/mi level (Ref. 8-11).

8.2.8 Noise

Engine noise, vibration, and harshness have been a problem in Ford's early stratified charge engine installations. Subsequently, vehicle and engine design modifications were incorporated, including insulation of the vehicle floor pan and stiffening of a number of structural members on both the engine and the vehicle. Although these changes resulted in a noticeable improvement, the noise and vibration characteristics of these vehicles remained unsatisfactory relative to standard engine/vehicle configurations (Ref. 8-9).

No information is available on the noise and vibration levels of the current Ford and Texaco SCE-powered vehicles.

8.2.9 Smoke and Odor

Smoke measurements conducted by Ford on single-cylinder stratified charge engines indicate exhaust opacities between 0.5 percent and 3 percent, compared with about 2 percent to 50 percent for diesels. In these tests, the smoke level had a tendency to increase with increasing EGR flow rate and engine load. A reduction in the smoke intensity could be accomplished by increasing the squish action in the combustion chamber to promote fuel mixing and vaporization (Ref. 8-9).

According to Ford (Ref. 8-9), the exhaust of its FCP engines has an upleasant diesel-like odor which, in high concentrations, can cause eye irritation. In general, the odor level is affected by the injection and spark timing, decreasing with engine throttling and increasing as the EGR flow rate increases (Ref. 8-9).

8.2.10 Driveability

During tests conducted by the EPA the emission-controlled M-151 vehicle incorporating Texaco's TCCS engine has shown poor driveability characteristics and insufficient power to negotiate the Federal Driving Cycle. These deficiencies are attributed entirely to the emission control system which was added to the engine to meet the 1976 statutory Federal emission standards (Ref. 8-15). Conversely, the uncontrolled M-151 TCCS vehicle has exhibited very good driveability (Ref. 8-8). According to Texaco, the cylinder injection results in rapid throttle response at all operating conditions and eliminates the warmup period generally required in carbureted engines. The uncontrolled engine also exhibits good starting and acceleration characteristics, even at temperatures below 10°F, without hesitation or stumble.

According to Ford, the driveability of its PROCO vehicle is good and engine stumble has never been encountered, even at very low engine speeds and full EGR flow. However, the acceleration performance of the emission controlled vehicle is impaired, and the idle quality is not quite satisfactory with the short-electrode spark plugs. The driveability of the vehicles utilizing Fast Burn engines was judged to be acceptable by Ford, with the exception of the Phase II engine at high EGR flow rates (Ref. 8-11).

8.2.11 Current Status

For the past several years, Texaco and Ford, under contract to the USA TACOM, have been involved in the development of stratified charge engines for potential application in the military L-141 light-duty engine. Without incorporation of emission-control equipment such as EGR, oxidation catalysts, and intake throttling, the HC and NO_x emissions from the L-141 stratified charge engines are comparable to conventional spark ignition engines, while CO is substantially lower. In this case, the fuel economy of the stratified charge engine powered vehicles is of the order of 30 percent higher than for equivalent conventional engines.

With emission control, the stratified charge engine powered vehicles tested to date over the Federal Driving Cycle meet the 1976 statutory emission standards at low mileage and show equal or slightly better fuel economy than the average 1973 certification vehicles tested at the same inertia weights. However, rapid deterioration of the emission control system performance is a major problem area.

The stratified charge engines manufactured to date by Texaco and Ford are hand-built experimental models, which are not suitable for mass production. Specifically, the current cylinder-head design is not compatible with mass-production techniques, and new production equipment and assembly procedures are required for the manufacture of the fuel injection and air fuel control systems.

A design study for a family of four, six, and eight-cylinder PROCO engines has been recently completed by Ford, with parts commonality being a prime consideration. Based on this study, Ford has concluded that the PROCO engine concept can be incorporated in these engine designs (Ref. 8-20).

8.2.12 Projected Status

Ford feels that limited production of PROCO engines could be feasible by 1977, providing the success experienced to date in the development of the PROCO engine continues (Ref. 8-11). However, before production commences, a number of potential problem areas must be further evaluated, including engine-component durability and emission control system deterioration characteristics, as well as fuel injection system requirements. In addition, several of the engine components require design modification to permit their manufacture by mass production techniques.

With respect to the Fast Burn concept, Ford plans to continue the development of the Phase II configuration. In particular, a new improved induction system is needed to reduce the HC and CO emissions. It appears that this concept might be sufficiently developed by 1977 to permit low-volume production.

Since oxidation catalysts are required on Ford's and Texaco's stratified charge engines to meet the 1975-76 statutory Federal emission standards, the need for unleaded gasoline would be unchanged by the adoption of these engine concepts. However, in view of the multifuel capability of the Texaco engine and its insensitivity to octane and cetane number, utilization of this engine would alleviate potential problems related to removal of lead from the fuel.

8.3 DIVIDED-CHAMBER STRATIFIED CHARGE ENGINES

8.3.1 General

Historically, the concept of a divided combustion chamber can be traced back to the first "oil engines" in the pre-diesel era. These engines were operated on heavy oils or naphtha injected into a spherical prechamber and vaporized in contact with the red-hot walls of the thermally insulated prechamber. Later, attempts were made to apply this concept to the spark ignition gasoline engines, as documented by over 100 patents. However, a practical and widespread application of the prechamber concept was realized only in the high-speed diesel engines, primarily due to the pioneering efforts of Ricardo in the early 1920s. The announcement in 1972 of the development of Honda's Compound Vortex Controlled Combustion (CVCC) engine marks the first practical prechamber spark ignition engine designed for use in automotive mass-production applications.

The combustion process in a divided-chamber engine proceeds in two stages. First, a rich air fuel mixture is ignited in the prechamber. Subsequently, the burning charge expands from the prechamber to the main combustion chamber, through communicating openings and ignites the lean air fuel mixture or the pure-air charge inducted into the main chamber. Several controlling factors are involved in this two-stage combustion process. These include the type of fuel and air-supply system utilized; the relative size of the prechamber to the main chamber; and the geometry of the prechamber, main chamber, and communicating passages.

Compared with open-chamber engines, the divided-chamber configurations have certain inherent disadvantages that are related to the higher surface-to-volume ratio of the combustion chamber and the higher flow-turbulence level during combustion. As a result, the heat losses to the cooling jacket and the emission of unburned hydrocarbons are increased. The net improvement in engine emissions and/or fuel economy depends on how well these effects are balanced by better combustion in the main chamber and improvement in engine cycle efficiency (for unthrottled engine operation).

Many domestic and foreign investigators and organizations are, or have been working on the concept of the divided-combustion-chamber gasoline engine. In addition to the most advanced development activity conducted by the Honda Motor Company, research and development work have been performed by the Ford Motor Company, Phillips Petroleum Company, Stanford University, University of Rochester, University of Wisconsin, Dynatech Corporation, and Volkswagen.

8.3.2 Power Plant Description

The divided-combustion-chamber designs currently known can be divided into three classes according to the ratio of the prechamber volume to the total compression-cylinder-clearance volume (prechamber, connecting passage, and main chamber): miniature prechambers, medium-size prechambers, and large prechambers. The design features of these engine categories are briefly described in the following paragraphs; discussions of selected engine designs are presented in Sections 8.3.2.1 through 8.3.2.3.

The miniature prechamber engines employ a small prechamber generally formed by an enclosed cavity around the spark plug electrodes of each cylinder. In these designs the prechamber volume is less than 10 percent of the total compression volume. The principal function of the miniature prechamber is to intensify the ignition and to stabilize the combustion process of the slightly lean air fuel mixture in each engine cylinder. This is achieved primarily by increased turbulence rather than stratification of the air fuel mixture. The effectiveness of this concept for emission reduction is rather moderate and might be attractive only in conjunction with other emission control methods, such as exhaust gas recirculation. This engine concept is exemplified by the Ford Torch engine and by the designs patented by Jozlin and Bishop.

The medium-size prechambers are the favored development targets of the various prechamber combustion engine concepts conceived to date (over 50 patents since 1910). These prechambers rarely exceed 25 percent of the total combustion chamber volume. The principal function of the medium-size prechamber is to (1) provide a means for effective stratification of the air fuel mixture, (2) facilitate ignition of the lean mixture in the main chamber, and (3) help generate the flow pattern and turbulence conducive to optimal combustion in the prechamber and in the main chamber. Engines in this category include the Honda CVCC engine and the designs by Broderson-Conta, Dynatech, Heintz, Nilov, Phillips, and Volkswagen.

The large-prechamber engine configuration represents a more recent concept which has been researched by Newhall and by the Ford Motor Company. In this design, the size of the prechamber ranges from about 50 to 85 percent of the clearance volume and communicates with the main chamber (secondary chamber) through a large orifice. The objective of the large prechamber is to provide for reliable stratification of the air fuel mixture. Similar to divided-chamber diesels, all fuel is injected into the prechamber. Upon ignition, the burning charge expands from the prechamber into the main chamber to continue the combustion process. It is hypothesized that the relatively cool air in the main chamber assists in quenching the NO formation reactions while providing additional oxygen required for the completion of oxidation reactions of carbon monoxide and hydrocarbons.

The aforementioned functions are not distinctly limited to only one size category. A certain overlap will result, depending upon the size and geometry of the prechamber and the prechamber fuel and air supply systems utilized.

8.3.2.1 Miniature Prechamber Engines

8.3.2.1.1 Ford Torch Ignition Engine

The Ford Torch Ignition Engine illustrated in Figure 8-15 features a torch chamber cavity or miniature prechamber located outside the main combustion chamber with which it is connected by one or more small holes. The carbureted air fuel mixture is introduced by a conventional manner into the engine cylinder, and part of it is transferred into the torch chamber during the compression stroke. The spark plug, located in the torch chamber, ignites the mixture in this chamber and "torches" of burning mixture are expelled through the torch chamber holes into the main combustion chamber. The torches of burning mixture penetrate deep into the main chamber, thus creating large area flame fronts that promote rapid completion of the lean-mixture combustion process.

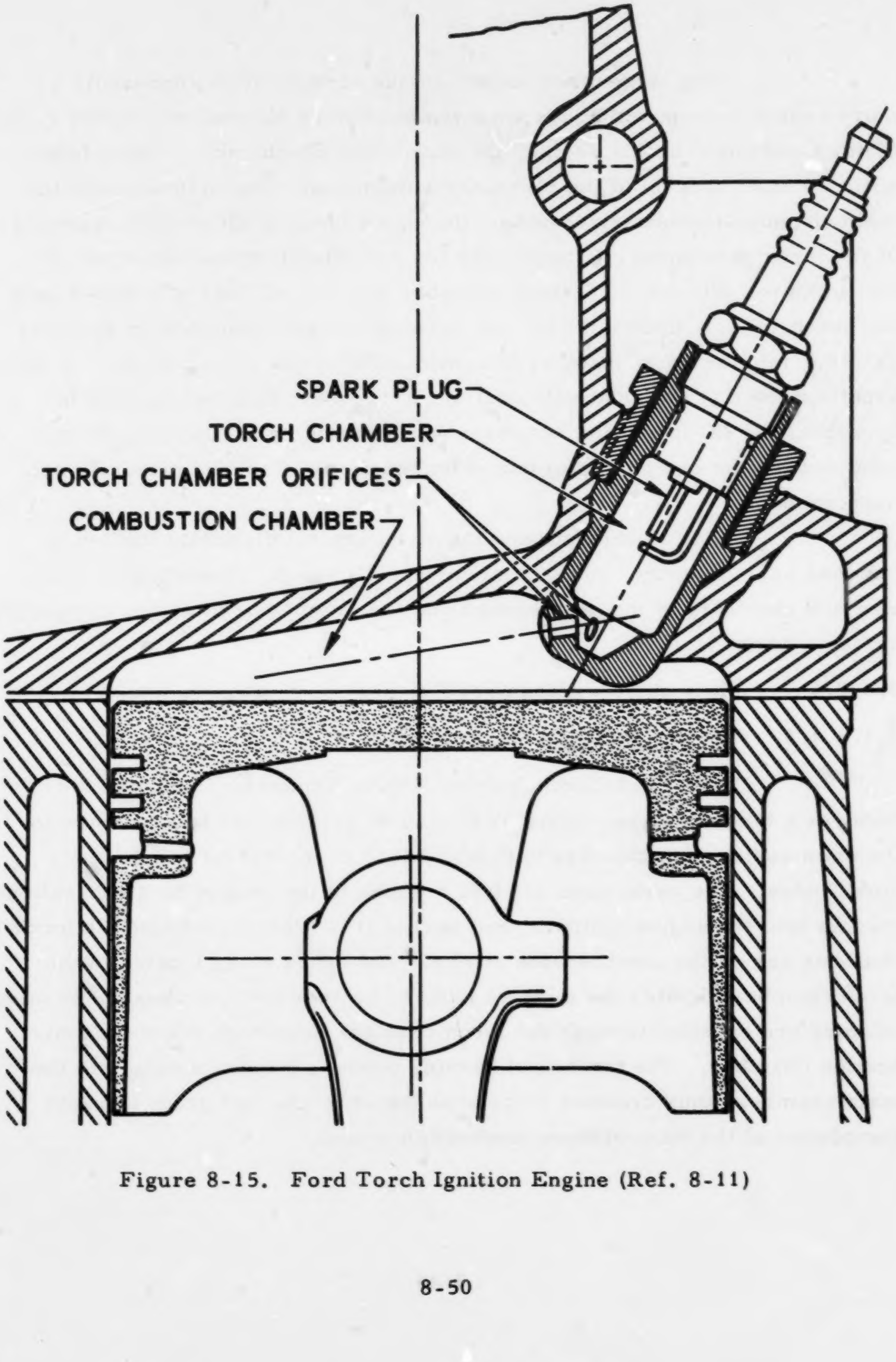

SPARK PLUG
TORCH CHAMBER
TORCH CHAMBER ORIFICES
COMBUSTION CHAMBER

Figure 8-15. Ford Torch Ignition Engine (Ref. 8-11)

Since there is no provision for enrichment of the combustible mixture in the torch chamber, the air fuel ratio of the engine is adjusted to about 15.5 (Ref. 8-11). The design changes with respect to conventional carbureted gasoline engines are limited to the engine head in the vicinity of the spark plug. No change in weight, volume, materials, or fuel requirement is indicated (Ref 8-11).

8.3.2.1.2 Other Concepts

A number of recent patents involving the concept of the miniature prechamber indicate current interest, research, and development in this engine configuration. Two of these concepts, which present new ideas for further functional improvement, are discussed in the following paragraphs.

Figure 8-16 presents the concept of a miniature prechamber equipped with a check valve according to a U. S. patent granted to Joseph A. Jozlin (Ref. 8-21). The objective of the check valve arrangement is to provide a sufficiently large port for the unobstructed entry of combustible mixture into the prechamber during the compression stroke and, subsequently, to block free exit of the burning mixture from the prechamber at the beginning of the piston power stroke. Upon ignition, high pressure is produced in the prechamber, and the burning mixture is then forced at high velocity through a set of connecting holes into the main combustion chamber. In an alternative embodiment, the check valve is replaced by a cam-operated poppet valve.

Figure 8-17 shows the miniature prechamber concept patented by I. N. Bishop, et al. (Ref. 8-22). In this case, direct fuel injection is employed instead of carburetion. The fuel injection nozzle is disposed to inject fuel into the main combustion chamber contiguous to the opening between the prechamber and the main chamber. The configuration of the main chamber and the positioning of the connecting hole (in the prechamber) are such that an air flow is set up from the main combustion chamber into

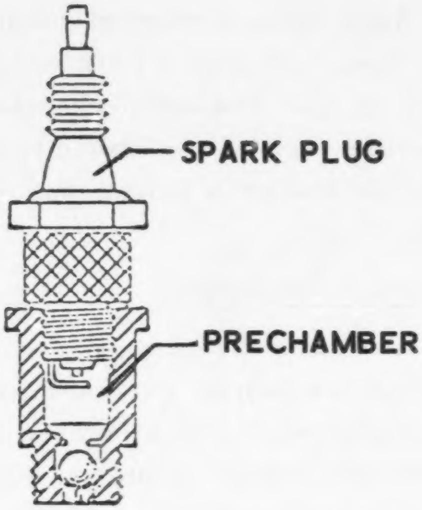

SPARK PLUG
PRECHAMBER

Figure 8-16. Jozlin Prechamber Concept(Ref. 8-21)

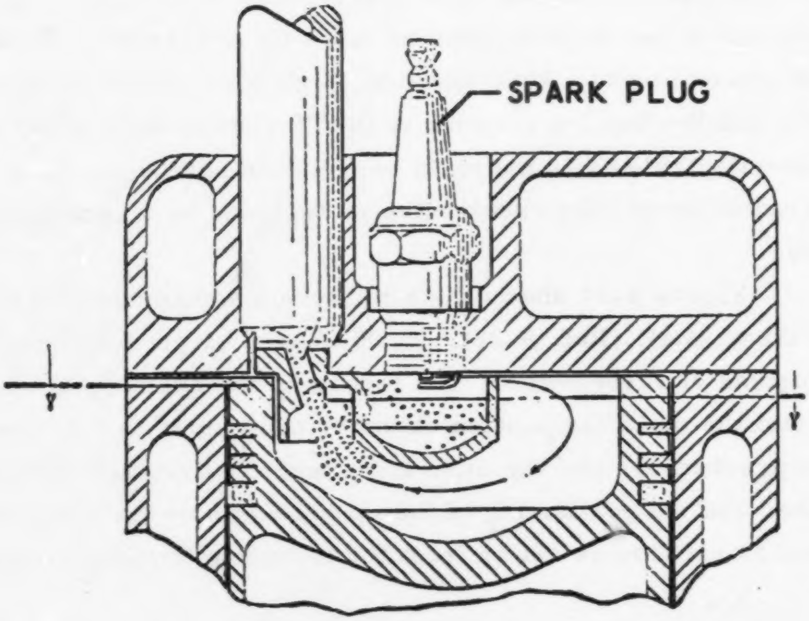

SPARK PLUG
Y
Y

Figure 8-17. Bishop Prechamber Concept (Ref. 8-22)

the prechamber during the compression stroke of the piston. The air flow sweeps a portion of the fuel droplets injected into the prechamber to provide a locally rich air fuel mixture. Since the mixture formed in the prechamber is richer than exists in the main chamber, it would appear that this concept has the potential of further extending the lean limit of the engine operation.

8.3.2.2 Medium Size Prechambers

8.3.2.2.1 Honda CVCC Engine

The original Honda CVCC (Compound Vortex Controlled Combustion) engine is an in-line, four-cylinder, four-stroke, 1948 cm^3 displacement (119 in^3) engine with a rating of 70 bhp at 5,000 rpm (Ref. 8-23). Prechamber volume is in the range of 10 to 20 percent of the total combustion chamber volume, and both parts of the chamber communicate through a small orifice. Figure 8-18 shows the principle of the design.

In this design, the smallest venturi of the three-barrel carburetor supplies a rich mixture to each prechamber. The other two venturis supply the main engine chambers with a very lean mixture. The main chamber utilizes a conventional inlet valve located in the chamber, while the mixture to the prechamber enters through a small inlet valve that opens into the prechamber. The prechamber inlet valve is actuated by means of an independent small rocker arm driven by the engine camshaft. The charge is stratified into three air fuel ratio regions: (1) a rich region in the prechamber, (2) a lean region in the main chamber, and (3) a region of moderate mixture strength in the vicinity of the orifice connecting both chambers. Combustion is initiated by the spark plug located in the prechamber.

According to Honda, combustion in the CVCC engine is controlled and proceeds at very slow rates to reduce the peak combustion temperature and the formation of NO. However, the temperatures are maintained high enough for a sufficiently long period of time during the expansion stroke to complete the oxidation of CO and HC. Conventional unmodified spark plugs are used on this engine. Apparently, optimum values exist for this engine with respect to configuration and size of the prechamber and the relative

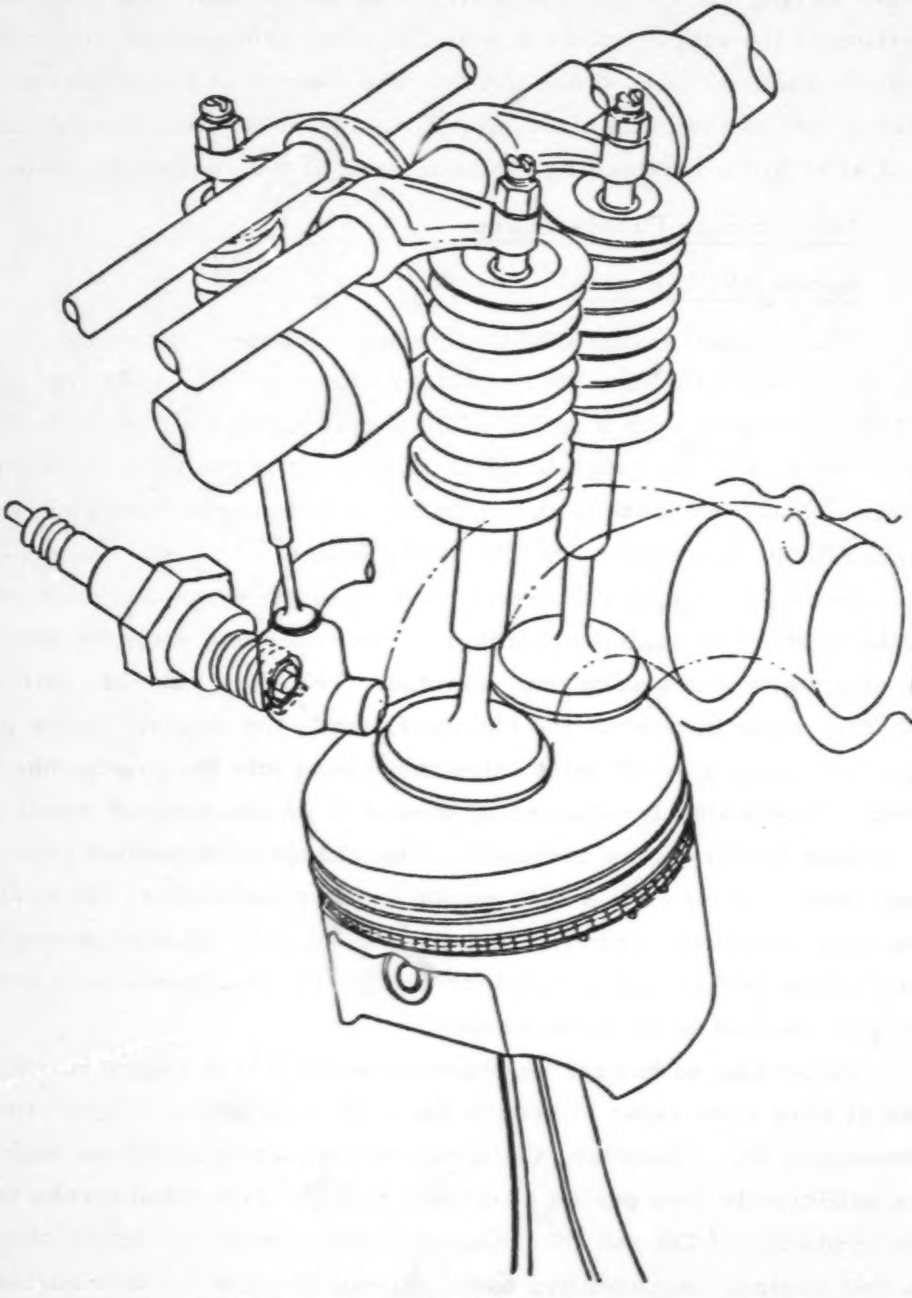


Figure 8-18. Honda CVCC Divided Chamber Stratified Charge Engine (Ref. 8-23)

positions of the auxiliary intake valve, spark plug, torch nozzle, and main combustion chamber. The air fuel ratios supplied to the prechamber and the main chamber vary with the operating conditions and are controlled with respect to prechamber scavenging, desired emission levels, and other performance parameters.

8.3.2.2.2 Broderson-Conta Engine (University of Rochester)

Figure 8-19 presents the concept of the Broderson-Conta engine (Ref. 8-24). Supposedly, this engine approches the diesel efficiency at light loads. In the engine, the fuel is directly injected into the prechamber while an unthrottled, carbureted, lean air fuel mixture is inducted into the main chamber. The engine load is varied by changing the amount of fuel supplied through the carburetor, and at very light loads the main chamber is charged with air only. Apparently, engine noise has been a persistent problem area.

8.3.2.2.3 Dynatech Prechamber Engine

A rather sophisticated approach was followed by Dynatech Corporation, which developed a prechamber concept for small, two-stroke engines (Ref. 8-25). In this design, fuel is injected at low pressures directly into both the prechamber and main chamber. The rate of fuel injection into the prechamber remains constant, while the main chamber-injection rate is varied with engine load. The injection system is electronically controlled, and air admission is unthrottled.

8.3.2.2.4 Ford Three-Valve-Prechamber Engine

Ford has conducted Three-Valve-Prechamber engine research and development work using both single-cylinder and V-8 engines (Ref. 8-11). In general, development-vehicle emission results have not reached the levels predicted from dynamometer tests. Major shortcomings of the present hardware include insufficient heating of the main and secondary induction systems, which results in high HC emissions due to excessive richness produced during vehicle deceleration. Vehicles equipped with current hardware can be operated satisfactorily with a fixed throttle angle in the secondary rich-mixture carburetor. No design information has been released by Ford.

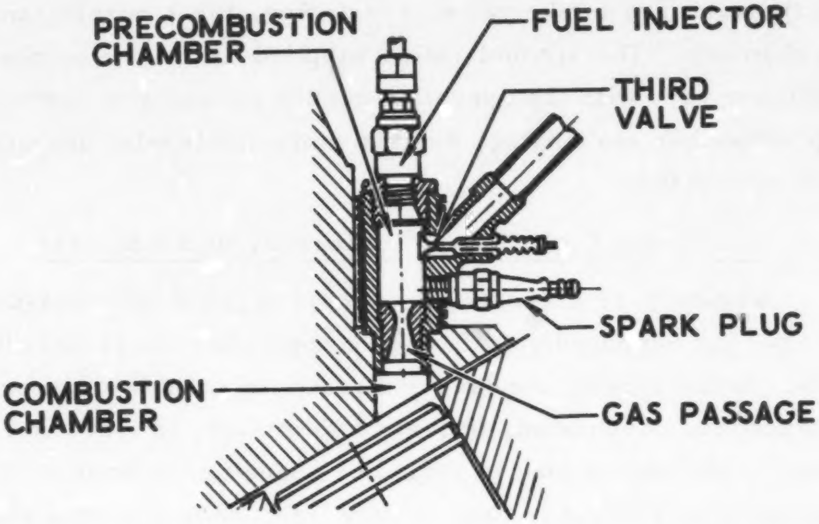


Figure 8-19. Broderson-Conta Engine (Ref. 8-24)

8.3.2.2.5 Heintz Ram-Straticharge Engine (Stanford University)

As illustrated in Figure 8-20, the Heintz engine is similar to previously discussed concepts although utilizing a more complex prechamber configuration (Ref. 8-26). Like the Honda CVCC design, the Heintz engine employs intake air throttling to limit the maximum air fuel ratio to values below 30 to assure complete combustion of the fuel. The original Heintz engine has since been modified and applied to both two and four-cycle engines. No data are available on the performance and emissions of these designs.

8.3.2.2.6 Nilov Engine

The basic concept of the Nilov engine is similar to the Honda CVCC system (Ref. 8-27). However, as shown in Figure 8-21, there appears to be a significant difference in the geometry of the main chamber and the location of the inlet valve. Also, the volume of the prechamber appears substantially smaller. Apparently, the Nilov engine has been developed to a high state of perfection and seems to be ready for production (Ref. 8-27).

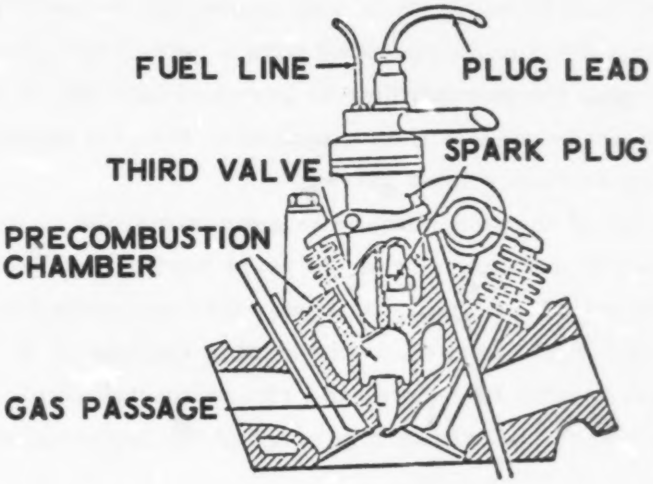

FUEL LINE
PLUG LEAD
SPARK PLUG
THIRD VALVE
PRECOMBUSTION CHAMBER
GAS PASSAGE

Figure 8-20. Heintz Ram Straticharge Engine (Ref. 8-26)

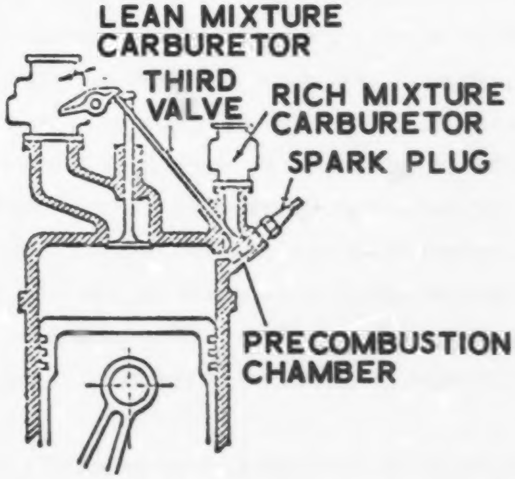

LEAN MIXTURE CARBURETOR
THIRD VALVE
RICH MIXTURE CARBURETOR
SPARK PLUG
PRECOMBUSTION CHAMBER

Figure 8-21. Nilov Engine (Ref. 8-27)

8.3.2.2.7 Phillips Petroleum Prechamber Engine

A comprehensive basic study was conducted by the Phillips Petroleum Company on a CFR engine equipped with a retrofit prechamber (Ref. 8-28). In this design, the prechamber is screwed into the standard spark plug hole, and the spark plug is then attached to the prechamber which is equipped with an auxiliary inlet valve.

The volume of the prechamber is approximately 13 percent of the total compression-clearance volume. In tests performed by Phillips at speeds between 1,000 and 1,800 rpm, the power was controlled by throttling the premixed air fuel charge into the engine cylinder. A very rich fuel air mixture, containing five times the stoichiometric fuel, is inducted into the prechamber at a rate of 2 percent of the main-chamber mixture flow.

8.3.2.2.8 Volkswagen Prechamber Engine

Volkswagen conducted considerable research and development on prechamber engines. Its latest development is illustrated in Figure 8-22. In this design, a spherical prechamber utilizing approximately 25 percent of the compression volume, is connected to the main combustion chamber by a flow-transfer passage. The main combustion chamber is disc shaped and contains no squish surfaces. The prechamber has no auxiliary valve. Fuel is injected into the prechamber by means of a low-pressure injection nozzle. The injection nozzle and spark plug are arranged in such a manner that a relatively rich mixture is present at the spark plug under all operating conditions. Mixture preparation is assisted by the shape of the prechamber, with the transfer passage entering almost tangentially. Load control is accomplished by the amount of carbureted mixture introduced into the main combustion chamber.

According to Volkswagen, fuel injection timing need not be varied over the entire load and speed range. This represents a considerable design simplification, compared with the mixture stratification process in

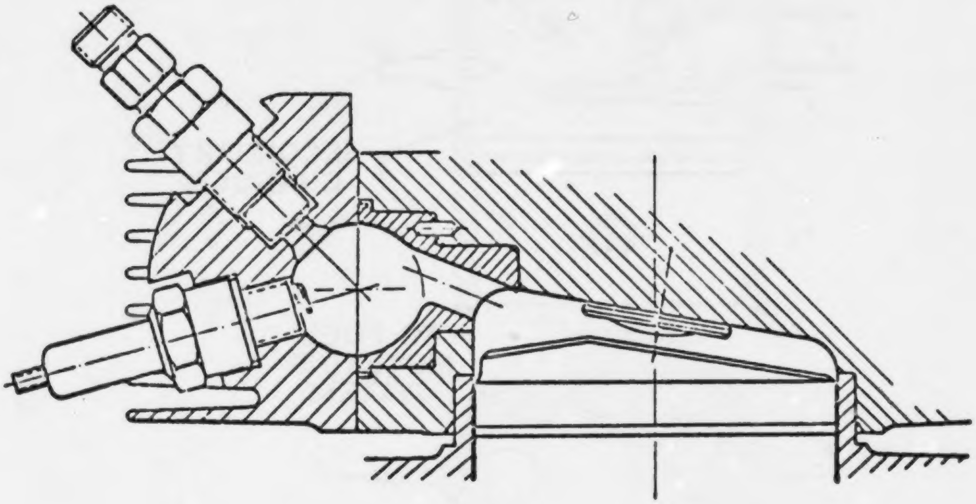


Figure 8-22. Volkswagen Prechamber Engine (Ref. 8-29)

an open combustion chamber, which generally requires the use of an injection timing control system. Initially, tests were conducted exclusively without intake air throttling. Later, partial throttling was employed in order to reduce the emissions at low engine loads.

8.3.2.3 Large Prechambers

8.3.2.3.1 Newhall Engine

The divided combustion chamber concept by Newhall and El-Messiri is shown in Figure 8-23. In this design, all fuel is injected into the primary chamber at an intermediate point during the compression stroke. The geometry of the combustion chamber and the injection timing are so arranged that no raw fuel enters the secondary chamber. The spark plug, which ignites toward the end of the compression stroke, is located in the primary chamber near the orifice connecting both chambers.

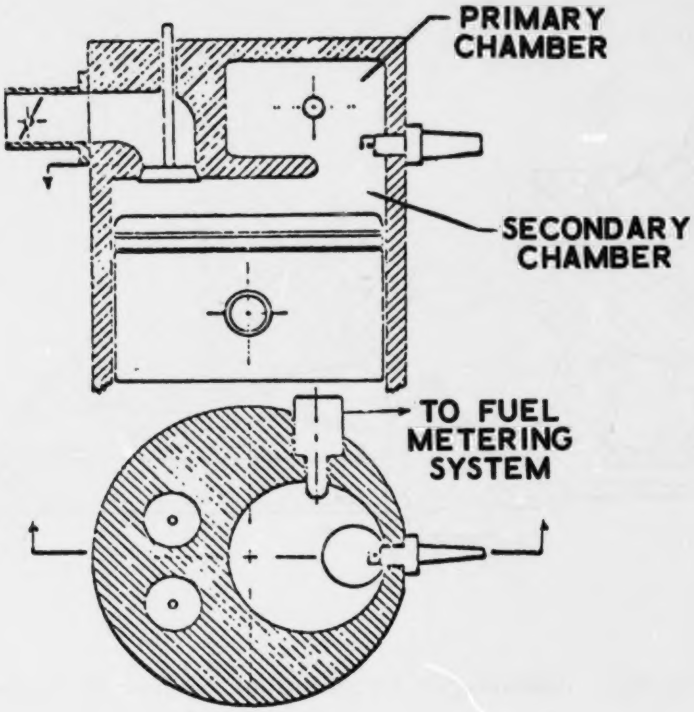


Figure 8-23. Newhall Prechamber Engine
(Ref. 8-30)

The power output of the engine is varied by conventional throttling. Air and fuel throttles are controlled simultaneously to maintain an air fuel mixture of appropriate strength in the primary chamber under all load conditions. Briefly, combustion in the Newhall engine proceeds as follows. Upon ignition of the prechamber mixture, combustion gases expand immediately into the secondary chamber, which initially contains relatively cool air. In principle, the rapid mixing and cooling process that ensues minimizes the formation of oxides of nitrogen, while at the same time the excess air contained in the secondary chamber promotes complete oxidation of hydrocarbons and carbon monoxide.

8.3.2.3.2 Ford Large Prechamber Engine

In addition to the work of Newhall, the Ford Motor Company has reported research and development work on large-volume prechamber engines (Ref. 8-11). Tests on a single-cylinder engine were conducted with and without EGR. With optimized adjustment under steady-state conditions, encouraging emission data were obtained. However, the operating range of the engine was limited to light-to-medium loads due to combustion harshness and poor air utilization.

In order to facilitate further development of this concept on multicylinder engines, a design effort has been initiated by Ford to incorporate this combustion concept into the 400 CID V8 engine. Dynamometer testing of this engine was scheduled to start in November 1973.

8.3.3 Performance

8.3.3.1 Emissions

8.3.3.1.1 Honda CVCC Engine

Evaluation of the emission characteristics of the Honda CVCC engine was conducted by the EPA (Refs. 8-31 and 8-32) and by the Honda Motor Company (Ref. 8-23). In the EPA test program, three Honda Civic cars equipped with CVCC engines were tested according to the 1975 Federal Test Procedure, using an inertia weight of 2,000 pounds. Two of the vehicles had odometer readings of about 1,500 miles, and the third car had just completed a 50,000-mile durability run. A summary of these emission tests is presented in Table 8-12, showing that the emissions from the vehicles were consistently below the 1975 Federal emission standards.

One of the three cars was also tested with an inertia weight of 3,000 pounds. The vehicle had adequate power to keep up with all the accelerations of the Federal Driving Cycle. Because of time constraints, this was a "hot start" test; i.e., the start of the test did not follow the standard 12-hour vehicle soak period but began shortly after the car had been

Table 8-12. Emissions and Fuel Economy, Honda CVCC Vehicles (Ref. 8-33)

Vehicle	Emissions, gm/mi, 1975 FTP			Fuel Economy, mi/gal	
	HC	CO	NO_x	1975 FTP	1972 FTP
Low mileage car (#3652. Five tests.)					
Average	0.18	2.12	0.89	22.1	21.0
Maximum[a]	0.21	2.28	1.05	22.4	21.5
Minimum[a]	0.15	1.96	0.75	21.9	20.6
Low mileage car (#3606. One test.)	0.23	2.00	1.03	20.7	19.5
50,000-mile car (#2034. Four tests.)					
Average	0.24	1.75	0.65	21.3	19.8
Maximum[a]	0.26	1.85	0.73	22.2	20.0
Minimum[a]	0.19	1.70	0.57	20.8	19.5

FTP = Federal Test Procedure (grams per mile)

[a] The values of the different columns do not necessarily correspond to the same test.

tested at 2,000-pound inertia weight. Although the initial period of the "cold start" was omitted, the test results appeared to show that the emissions of a Honda vehicle of 3,000 pounds, equipped with the same CVCC engine, would be lower than the statutory 1975 emission standards (Ref. 8-31).

Particulate emission tests conducted by Dow Chemical Company under EPA contract indicate that the particulate emissions from the CVCC vehicles are comparable to those from convenient engines using equivalent fuels.

To date, Honda has conducted emission durability tests on seven 2.0-liter CVCC vehicles accumulating a total of 437,000 miles. Based on these tests, the deterioration factors are approximately 1.10 to 1.15 for HC, 1.05 to 1.10 for CO, and 1.00 to 1.05 for NO_x, showing good emission control durability of the CVCC engines.

Honda has adapted the CVCC process to larger engines to demonstrate the feasibility of the concept in full-size American cars. Table 8-13 presents Honda's emission test results for a Chevrolet Vega car equipped with a 140 CID CVCC engine. As indicated, the emissions are below the statutory 1975 standards, and the fuel economy is slightly better than for the non-modified Vega.

Emission data obtained by the EPA for a CVCC modified 350 CID Chevrolet Impala automobile are listed in Table 8-14 (Ref. 8-32). Tests 1 and 4 demonstrate that CO and HC emissions below the statutory 1975-76 levels can be achieved with this vehicle, and that oxides of nitrogen levels are consistently below the 1976 interim standard of 2.0 grams per mile. During tests 2 and 3, problems associated with the hot soak period resulted in high levels of CO and HC during the bag 3 portion of the tests. The high CO level in bag 3 of test 2 appeared to be associated with flooding of the prechamber carburetor. This was corrected by Honda technicians prior to test 3. The high hydrocarbon level in bag 3 of test 3 was apparently caused by a false hot start which resulted in excessive cranking by the EPA driver. Another false start occurred on test 4, but the more experienced Honda driver was able to recover with less hydrocarbon penalty. It is believed that the hot start problems experienced on the prototype vehicle can be corrected with further engineering effort.

The consistently low cold start (bag 1) hydrocarbon and CO levels during all of the tests should be emphasized. An average value for bag 1 of 0.38 gm/mi HC and 3.51 gm/mi CO was observed. Good control of oxides of nitrogen was also demonstrated with this vehicle. An average value equal to 1.72 gm/mi was obtained for the 1975 FTP testing.

Table 8-13. Low-Mileage Emissions and Fuel Economy of Vega Vehicles Modified for CVCC (Ref. 8-23)

	Test Date	Emissions, gm/mi			Fuel Economy mi/gal(2)	No. of Tests
		HC(1)	CO(1)	NO_x(1)		
CVCC - 140 CID	9-72	0.26	2.9	1.2	17.9	2
CVCC - 140 CID, improved	11-72	0.26	2.6	1.2	18.6	2
CVCC - 140 CID, improved	4-73	0.22	2.4	1.2	18.9	3
Original 140 CID	-	2.1	10.6	3.8	17.2	2

(1) 1975 FTP
(2) 1972 FTP

Table 8-14. Low-Mileage 1975 FTP Emissions and Fuel Economy of Impala Vehicles Modified for CVCC (Ref. 8-32)

Test No.	HC (gm/mi)	CO (gm/mi)	NO_x (gm/mi)	Fuel Consumption (mi/gal)
1	0.27	2.88	1.72	10.5
2	0.23	5.01	1.95	11.2
3	0.80	2.64	1.51	10.8
4	0.32	2.79	1.68	10.2

Note: Above tests run at 5,000-pound inertia and 14.7 rear wheel hp at 50 mph (hp includes 10 percent for air conditioning)

8.3.3.1.2 Other Concepts

The emission data available for the other prechamber engines are briefly discussed in the following paragraphs.

Hot-start emission and fuel economy data published by Ford for its Torch Ignition Engine are presented in Table 8-15 (Ref. 8-11). The test vehicle was a 4,500-pound 1972 Gran Torino Sport equipped with an 8.5 compression ratio, 351-CID engine, and modified for torch ignition. Unfortunately, no baseline emission data from an unmodified engine are shown to indicate the degree of emission reduction attained. The NO_x levels are interesting; however, the relatively low CO values may be due primarily to the air injection into the exhaust manifold used in conjunction with the torch ignition concept. All reported tests were run with EGR, and the NO_x and CO emissions are typical of the levels obtained from conventional engines equipped with EGR and with lean carburetor settings.

Hot-start emission data obtained by Ford for a 1973 LTD and a 1973 Gran Torino equipped with 400-CID Three-Valve-Prechamber Engines are presented in Table 8-16. As indicated, CO is very low and comparable to the Honda CVCC data, but HC is much higher. Because of induction system heating problems, only hot-start tests were performed by Ford on these vehicles. Preliminary emission data taken by Ford on a single-cylinder experimental engine incorporating a large volume prechamber are listed in Table 8-17.

The Dynatech concept was tested by the EPA in a Saab vehicle in accordance with the 1975 FTP (Ref. 8-25). In these tests, the emissions were 6.4 g/mi HC, 3.6 g/mi CO, and 0.3 g/mi NO_x. The HC emissions are quite high and exceed the 1976 standards by an order of magnitude. Conversely, the NO_x level appears to be safely below the standard, while CO approaches the standard. Since JP-4 was used in these tests, there is some question as to the effect of this fuel type on the emission results.

The effect of the Phillips Petroleum retrofit prechamber on the emissions of an experimental CFR engine is presented in Figure 8-24.

Table 8-15. Hot-Start Emissions and Fuel Economy, Ford Gran Torino with Torch Ignition Engine (Ref. 8-11)

Test No.	Emissions, gm/mi			Fuel[1] Economy (mi/gal)	% EGR	Remarks
	HC	CO	NO_x			
5	2.78	8.75	2.22	12.25	7 - 15	15.5:1 A/F TDC timing
8	3.13	5.67	1.57	11.55	8 - 15	15.5:1 A/F w/air injection TDC timing
10	2.72	5.73	2.26	11.73	8 - 16	15.5:1 A/F 6° BTDC timing w/air injection
12	1.86	0.64	1.27	11.76	5 - 15	15.5:1 A/F TDC timing w/air injection; 1 hole torch
13	2.00	13.87	1.26	11.71	6 - 15	15.5:1 A/F TDC timing w/air injection; 1 hole torch

[1] Carbon balance method; Federal Driving Cycle

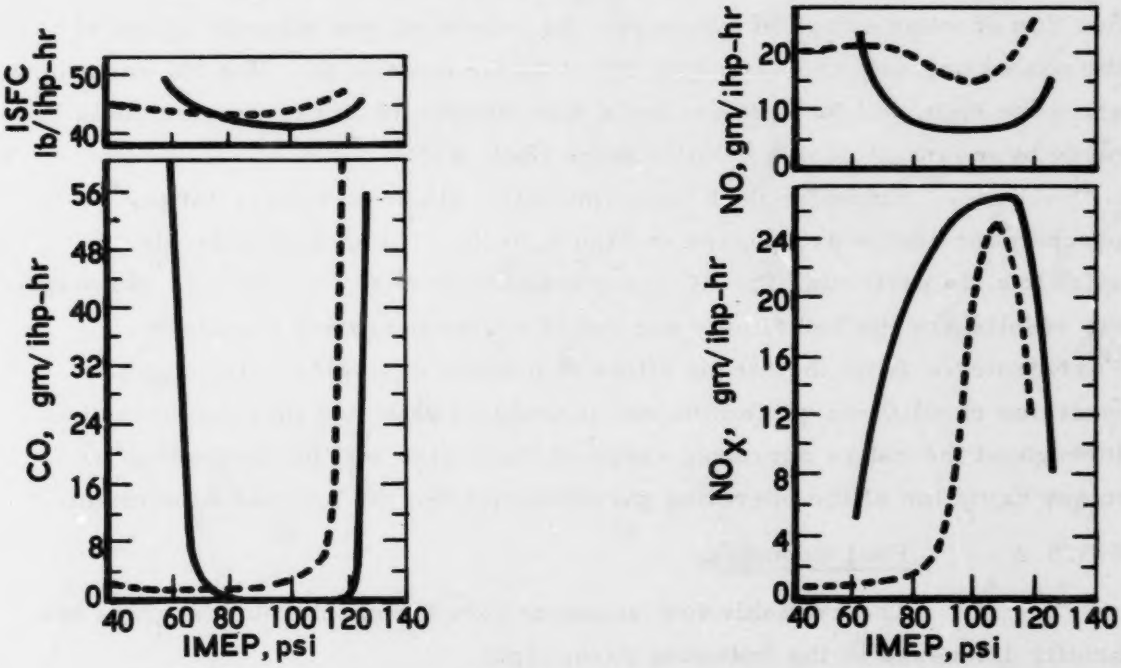
Table 8-16. Ford Vehicle Hot-Start Emissions and Fuel Economy, Three-Valve-Prechamber Engines, 1975 FTP(Hot) (Ref. 8-11)

Test Vehicle	Engine	Emissions, gm/mi			Fuel Economy (mi/gal)	EGR Flow
		HC	CO	NO_x		
1973 LTD*	400 CID Phase 1	1.78	3.36	1.29	9.8	0
1973 Gran Torino*	400 CID Phase 2	2.40	3.92	1.25	11.0	0

*Inertia weight 4,500 lb

Table 8-17. Ford Large-Prechamber Emissions and Fuel Economy, Single-Cylinder Engine (Ref. 8-11)

Speed (rpm)	Indicated Mean Effect. Pressure, psi	Emissions, gr/ihp-hr			Indicated Fuel Consumption (lb/ihp-hr)
		HC	CO	NO_x	
1500	40	1.2	5.0	0.5	0.395
1500	70	1.1	6.0	0.5	0.377



Note: ——— Standard Engine (CR = 6.0)

- - - Prechamber Engine (CR = 6.46)

1000 rpm
Maximum Power Spark Timing

Figure 8-24. Phillips Single-Cylinder Prechamber Engine Emissions and Indicated Specific Fuel Consumption vs Indicated Mean Effective Pressure (IMEP) (Ref. 8-28)

The prechamber-equipped engine produces less NO_x over the entire range of engine load (IMEP). NO_x emission was very low for engine loads below 70-psi IMEP. The comparison with emissions from an equivalent standard engine demonstrates the potential of the prechamber engine for reducing NO_x emissions. However, emissions of HC are higher in this prechamber engine at loads greater than 70-psi IMEP and markedly lower for engine loads below 60-psi IMEP. A similar trend was observed for CO emission. Reference 8-27 presents emission test results for a wide range of engine loads, rpm, air fuel equivalence ratios, and percent of throttle opening.

Steady-state emissions published by Volkswagen for its single-cylinder prechamber test engine are presented in Figure 8-25 as a function of mean effective pressure. As indicated, the size and shape of the connecting orifice have some effect on the emissions. The HC emissions are quite high, but Volkswagen feels that significant improvements could be made by means of design modifications (Ref. 8-26).

Emission data from Newhall's single-cylinder, large-prechamber engine are plotted in Figure 8-26. The emission levels are quite low, in particular the HC at equivalence ratios of about 0.8. However, the results are limited to only one set of engine operating conditions. Furthermore, from the strong effect of fuel air equivalence ratio on the emission of all three contaminants, it could be expected that the emissions throughout the entire operating range of the engine will be very sensitive to any deviation of the operating parameters from the optimal adjustment.

8.3.3.2 Fuel Economy

The available fuel economy data for prechamber engines are briefly discussed in the following paragraphs.

Average, maximum, and minimum fuel economy data reported by Honda for its CVCC-powered test vehicles are listed in Table 8-12. As shown, the average 1975 FTP fuel economy of the three Honda vehicles tested (21.3 mpg) was 16 percent lower than the average of the 2,000-pound 1973 certification vehicles tested by EPA (25.5 mpg). Honda data comparing

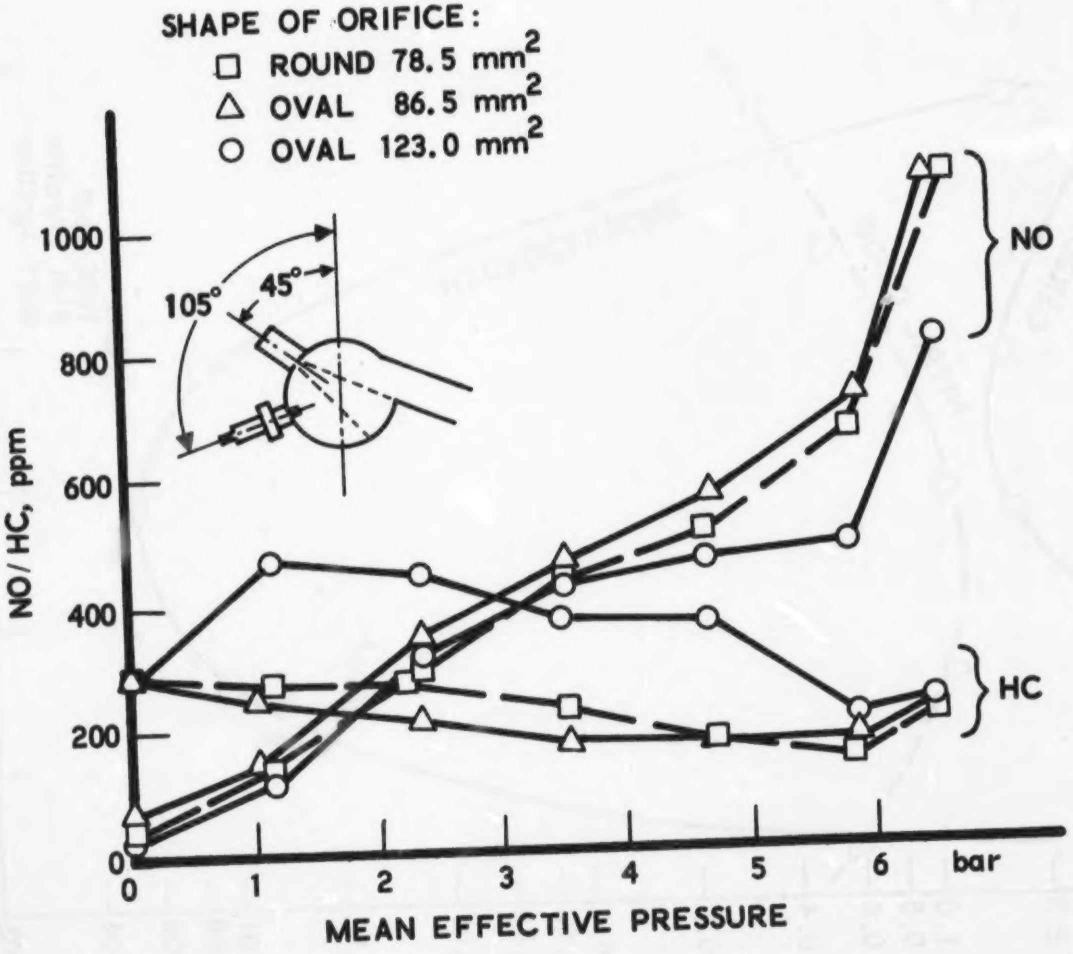


Figure 8-25. Emission Characteristics, Volkswagen Single-Cylinder Prechamber Engine (Ref. 8-29)

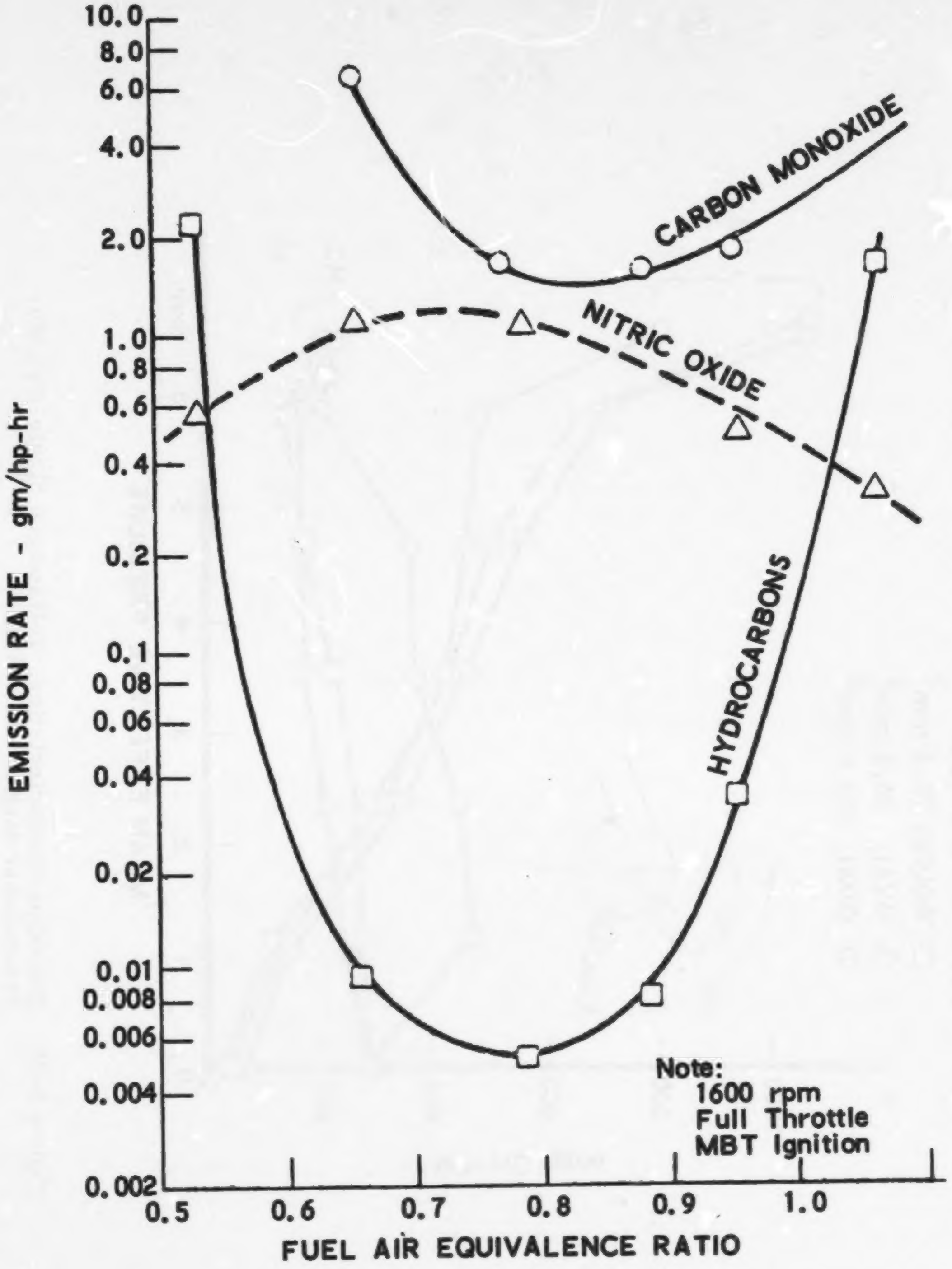


Figure 8-26. Preliminary Emission Data, Newhall Prechamber Engine (Ref. 8-33)

the CVCC-powered Civic to the standard Civic with a conventional engine also show about 10 percent fuel economy penalty for the CVCC version. However, the fuel economy of this car at 3,000-pound inertia weight (18.7 mpg) compared quite favorably with both of the average 3,000-pound 1973 certification vehicles (16.2 mpg). Comparison of the data in Tables 8-13 and 8-14 indicate slightly better fuel economy for the CVCC-modified Vega and Impala vehicles.

Fuel economy data from Ford's 1972 Gran Torino vehicle equipped with a Torch Ignition Engine and EGR are listed in Table 8-15. As indicated, the average fuel economy of this vehicle operated with different EGR rates and spark timing is about 10 percent better than that of the average 1972 certification vehicle tested at 4,500-pound inertia weight.

Fuel economy data for Ford's Three-Valve Prechamber Engine installed in a 1973 LTD and a 1973 Gran Torino vehicle are listed in Table 8-16. As indicated, the fuel economy of the two vehicles is slightly better than for the corresponding 1973 certification vehicles.

Limited indicated specific fuel consumption (ISFC) data for Ford's single-cylinder large prechamber engine were presented in Table 8-17. The data are in reasonable agreement with the fuel consumption data reported by Newhall for a similar type engine (Ref. 8-30).

The specific fuel consumption data plotted in Figure 8-24 for the Phillips prechamber engine indicate better economy relative to the standard CFR engine for indicated mean effective pressures (IMEP) below about 70 psi. Conversely, above 70 psi, the standard engine is better. Considering the low power requirements in urban traffic, this result suggests the possibility of substantial fuel savings that might be realized with the prechamber engine.

8.3.4 Noise and Odor

Engine noise and roughness of operation tend to be more severe in prechamber engines than in conventional spark ignition engines. This is particularly noticeable at higher engine loads. Some success in noise suppression was achieved by modifications in the geometry of the communicating passages between the prechamber and the main chamber (Ref. 8-23). Currently, it is not clear whether the engine noise and roughness are due to high pressure rise rates in the engine cylinder or some secondary gas dynamic effects in the main chamber.

There appears to be disagreement regarding the exhaust odor characteristics of prechamber engines. Honda (Ref. 8-23) reports aldehyde levels below 0.03 grams per mile for CVCC-powered vehicles, much lower than normally found in conventional spark ignition engines. Test data published by Wimmer (Ref. 8-28) indicate that the aldehyde emissions in prechamber engines increase with increasing overall air fuel ratio.

8.3.5 Maintenance Requirements

No special maintenance problems or needs have been identified with any of the prechamber engine concepts presently in development. Honda, after extensive durability tests, concluded that the maintenance procedures required for CVCC engines are no different from those for conventional spark ignition engines, and the emissions can be preserved within the scope of ordinary maintenance procedures.

8.3.6 Driveability

The only information on vehicle driveability effects related to the incorporation of a prechamber engine was reported by the EPA. Apparently, no problems were encountered during road performance evaluation of the CVCC vehicles; the engines were very responsive and showed good acceleration capability (Ref. 8-31).

8.3.7 Current Status

The development status of the previously discussed spark ignition prechamber engine concepts is summarized in Table 8-18. To date, the Honda CVCC engine is the only prechamber engine which has been subjected to extensive performance tests and evaluations and which has achieved production prototype status. Apparently, a number of the other concepts in

Table 8-18. Current Prechamber Engine Development Status

Prechamber Type		R&D Status				Performance								
		Single-Cylinder Experimental	Multi-Cylinder Experimental	Production Prototype	Mass Production	Emission of NO_x	Emission of HC	Fuel Consumption	Odor	Noise	Driveability Problems	Power Loss	Octane Requirement	Operational Cost Factor
Miniature	Ford Torch Engine	P	I	O	N	-3	-3	N	N	N	N	N	N	N
	Jozlin Prechamber	N	N	N	N	N	N	N	N	N	N	N	N	N
	Bishop Precham. (Ford)	N	N	N	N	N	N	N	N	N	N	N	N	N
Medium	Wimmer (Phillips Petroleum)	A	N	N	O	-5	-5*	0	3	2	0	1	-2	N
	Honda CVCC Engine	N	C	C	O	-5	-5	2	-3	0	0	0	N	2
	Nilov Engine	N	N	N	N	N	N	N	N	N	N	N	N	N
	Broderson-Conta (U. of Rochest.)	A	N	N	N	N	N	-3	N	3	N	N	N	N
	Heintz Ram-straticharge	A	A	N	O	N	N	N	N	N	N	N	N	N
	Dynatech Corp. (2-stroke)	N	A	N	O	-5	0	N	N	N	N	N	N	N
	Ford (3 valve prechamber)	A	P	N	O	-4	-4	1	N	N	N	N	N	N
	Volkswagen Prechamber Engine	P	I	O	O	N	N	N	N	N	N	N	N	N
Large	Newhall (Univ. of Wiscons.)	P	N	O	O	-5	-5	0	N	N	N	N	N	N
	Ford (Large Prechamber)	P	I	N	O	-4	-4	N	N	3	N	3	N	N

*extremely low base emission

N	no data available
O	not initiated as yet
I	initiated
P	in progress
A	advanced development
C	completed

3	substantially increased	(< 30%)
2	increased	(< 10%)
1	slightly increased	(< 5%)
0	no change vs. standard	
-1	slightly decreased	(< 5%)
-2	decreased	(< 10%)
-3	substantially decreased	(< 30%)
-4	strongly decreased	(> 30%)
-5	radically decreased	(> 80%)

the medium-size category also have reached an advanced state of development. Conversely, little concrete information is available relative to the status in research and development of small and large prechamber engines. Based on the technical information available, the following observations can be made:

a. A substantial reduction of the NO_x, HC, and CO emissions is reported for the medium-size prechamber concepts discussed in this report.

b. In several cases, the fuel consumption of the prechamber engine is higher than that of equivalent conventional spark ignition engines. However, this may be compensated for to some extent by an attendant reduction in fuel-octane requirement.

c. Relative to conventional spark ignition engines, prechamber engines tend to have higher noise levels and may suffer some loss in power.

d. The production cost of prechamber engines is expected to be slightly higher than for conventional engines.

e. No driveability problems are indicated.

f. Good engine durability and no unusual maintenance problems and expenses are expected for mass-produced prechamber engines.

8.3.8 Projected Status

The trends noted above may change significantly if unthrottled engine operation can be achieved in any of the discussed concepts. In addition to lowering emissions, this could result in an appreciable reduction in fuel consumption.

Two of the reported concepts (Broderson-Conta and Dynatech Corporation) are designed for unthrottled operation. However, the lack of emission and performance data precludes the assessment of the feasibility of these engines at this time.

Even if future research should show that the condition of unthrottled operation cannot be satisfied, it might be possible to develop an optimal prechamber concept which would achieve a substantial reduction in emissions without appreciably increasing fuel consumption. In exploring this possibility, the following considerations may prove to be valid:

a. Miniature Prechambers. With decreasing size of the prechamber, the engine characteristics approach the characteristics of a standard engine. As a result, the potential for emission reduction decreases, but the fuel economy penalty decreases also.

b. Large Prechambers. With increasing size of the prechamber, the engine characteristics approach the characteristics of an open-chamber stratified charge engine. The potential for emission reduction increases; however, the typical problems associated with open-chamber stratified charge engines become more pronounced.

c. Medium Size Prechambers

   1. The benefits of emission reduction and fuel economy improvement might be realized at low engine loads only. At high engine loads, these trends might be reversed.

   2. Achievement of an optimal engine system may require development of more sophisticated prechamber concepts.

8.4 CONCLUDING REMARKS

To date, no information has been published on studies related to two-stage combustion in prechamber gasoline engines. Hypotheses on the kinetics of NO_x formation in the prechamber, transition zone, and main chamber have been formulated (Ref. 8-30), and certain flow patterns in the prechamber and main chamber have been assumed by most investigators (Ref. 8-23). However, these hypotheses and assumptions have not been substantiated by theoretical and/or experimental investigations. In reality, the stratification pattern anticipated from a given chamber design may be distorted by gas dynamic (e.g., resonance) effects in both prechamber and main combustion chamber to such a degree that the optimal combustion conditions are not attained (Refs. 8-34 and 8-35). These effects may also be associated with the noise and roughness of prechamber-engine operation. To elucidate complex behavior of this kind, comprehensive basic studies appear indispensable.

The conclusive assessment of the potential of the gasoline engine prechamber concept for offering a satisfactory solution to low emissions and acceptable fuel economy may depend on the advancement of understanding of the principal gas kinetic and gas dynamic process involved. Moreover, this may provide a basis for successful development of an unthrottled prechamber engine.

APPENDIX A

ENVIRONMENTAL PROTECTION AGENCY

PROTOTYPE VEHICLE PERFORMANCE SPECIFICATION
FULL-SIZE PASSENGER CAR

January 3, 1972

Division of Advanced Automotive
Power Systems Development

2929 Plymouth Road
Ann Arbor, Michigan 48105

Approved: George M. Thur
Mr. George M. Thur
Chief, Power Systems Branch

ADVANCED AUTOMOTIVE POWER SYSTEMS (AAPS)

PROTOTYPE VEHICLE PERFORMANCE SPECIFICATION[*]

January 3, 1972

The AAPS Vehicle performance design specification presented below is intended to provide:

A common objective for prospective contractors.

Criteria for evaluating proposals and selecting a contractor.

Criteria for evaluating competitive power systems for entering first generation system hardware.

Advisory criteria to assist the contractor in such areas as rolling resistance, vehicle air drag etc.

The derived criteria are based on typical characteristics of the class of passenger automobiles with the largest market volume produced in the U.S. during the model years 1969 and 1970. It is noted that emissions, volume and most weight characteristics presented are maximum values while the performance characteristics are intended as minimum values. Contractors and prospective contractors who take exceptions must justify these exceptions and relate these exceptions to the technical goals presented herein.

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[*] Supersedes "Vehicle Design Goals - Six Passenger Automobile"
(Revision C - May 28, 1971)

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INTRODUCTION

The design of an automobile from a total systems standpoint could be expected to result in major benefits in cost, safety, and performance. This specification is intended as a step along that path, describing a propulsion system that can be installed into engine compartments as they now exist. Integration of the vehicle accessories within the propulsion system are highly desirable. Following the successful demonstration of this development further optimization of the propulsion system with the power train, suspension, and vehicle styling would be possible.

1. VEHICLE WEIGHT WITHOUT PROPULSION SYSTEM - W_o

W_o is the weight of the vehicle excluding the propulsion system. This weight includes, but is not limited to the frame, body, glass, trim, suspension, wheels (rims and tires), service brakes, seats, upholstery, sound absorbing materials, insulation, dashboard instruments, accessory ducting and wiring, accessories, and all other components not included as part of the propulsion system.

Accessories are defined as driver assistance and passenger convenience components and subsystems not essential to propulsion system operation. Included are power steering systems, power brake systems and passenger compartment heating and air conditioning systems.

W_o is fixed at 2700 lbs.

2. PROPULSION SYSTEM WEIGHT - W_p

W_p includes the energy storage subsystem (including fuel, containment, and supply and deliver ducting), power conversion subsystem (including auxiliaries and control) and power transmitting subsystem (including transmission and drive train to the driven wheels).

Auxiliaries are defined as components and subsystems essential to the operation of the power conversion system. Included are electric power generating subsystems, starting subsystems, exhaust subsystems, motors fans, flowers, pumps, and fluids.

Lightweight propulsion system are highly desirable, however, the maximum allowable propulsion system weight, W_{pm}, is 1500 lbs.

3. VEHICLE CURB WEIGHT - W_c

$W_c = W_o + W_p$.

The maximum allowable vehicle curb weight, W_{cm}, is 4200 lbs. (2700 + 1500 max. = 4200).

4. VEHICLE TEST WEIGHT - W_t

$W_t = W_c + 400$ lbs. W_t is the vehicle weight at which all accelerative maneuvers, fuel economy and emissions are to be calculated. (Items 9c, 11c, 11d, 11e, 11f, 12, 13).

The maximum allowable test weight, W_{tm}, is 4600 lbs. (2700 + 1500 max. + 400 = 4600).

5. GROSS VEHICLE WEIGHT - W_g

$W_g = W_c + 1100$ lbs. W_g is the gross weight vehicle weight at which sustained velocity capability at 5 percent and 30% grades is to be calculated (Item 11f). The 1100 lbs. load simulates a full load of passengers and baggage.

The maximum allowable gross vehicle weight, W_{gm}, is 5300 lbs. (2700 + 1500 max. + 1100 = 5300).

6. PROPULSION SYSTEM VOLUME - V_p

V_p is the volume allotted for all items identified under item 2. The propulsion system shall be packagable in such a way that the volume encroachment on either the passenger or luggage compartment does not exceed the following:

a) The transmission tunnel may not be widened so as to decrease the selected production vehicle clearance between the accelerator pedal and the tunnel. The accelerator pedal may not be relocated.

b) Intrusion of the tunnel into the passenger side of the vehicle may be increased by a maximum of 1.5 inches.

c) The tunnel height may be increased by a maximum of 2 inches but without affecting the full fore and aft adjustment of the front seat of the vehicle. The front seat may not be raised.

The propulsion system shall not violate the vehicle ground clearance lines as established by the manufacturer of the vehicle used for propulsion system/vehicle packaging. Additionally, the propulsion system shall not violate the space allocated for wheel jounce motions and vehicle steering clearances. Necessary external appearance (styling) changes will be minor in nature. The propulsion system shall also be packagable in such a way that the handling characteristics of the vehicle are not degraded.

7. AIR DRAG

The product of the drag coefficient, C_d, and the frontal area, A_f, is to be used in air drag calculations. The product of C_dA_f has a value of 12 ft^2. The air density used in computations shall correspond to the applicable ambient air temperature.

8. ROLLING RESISTANCE

Rolling resistance, R, is expressed in the equation (R = (W/65 $[1 + (1.4 \times 10^{-3}V) + (1.2 \times 10^{-5}V^2)]$ lbs. V is the vehicle velocity in ft/sec. W is the vehicle weight in lbs.

9. PROPULSION SYSTEM EMISSIONS

The vehicle is to be tested for emissions in accordance with the procedure in the July 2, 1971 Federal Register and as further described in the Code of Federal Regulations Title 40 Part 85 for model year 1976 light duty vehicles (CFR 40-85). The vehicle test weight shall be W_t and the accessory load as defined in Section 14a. Ambient conditions are 14.7 psia and 85°F. Emission tests will be run with fuel specified in Section 10.

The Federal emissions standards are:

Hydrocarbons - 0.41 grams/mile maximum
Carbon Monoxide - 3.40 grams/mile maximum
Oxides of Nitrogen* - 0.40 grams/mile maximum

Prototype vehicles are to meet the following emissions goals to allow for production tolerances and life degradation. Measurement of emissions is to be taken after the system has operated for 100 hours.

Hydrocarbons - 0.20 grams/mile maximum
Carbon Monoxide - 1.70 grams/mile maximum
Oxides of Nitrogen* - 0.20 grams/mile maximum

Production of smoke, odors, aldehydes, ammonia, particulates or other undesirable emissions not now specified in the July 2, 1971 Federal Register are undesirable.

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*Oxides of nitrogen are to be measured or computed as NO_2.

10. FUEL

Emission tests will use the fuel specified below, however, the power system shall have the capability of meeting emission levels using commercially available unleaded fuels.

Item	ASTM Designation	Specification
Octane, Research, min.	D1656	91-93
Pb. (Organic), gm/U.S. gal	D 525	<0.02
Distillation range	D 86	-
I.B.P., °F	-	100-115
10 percent point, °F	-	140-150
50 percent point, °F	-	240-250
90 percent point, °F	-	330-340
E.P. °F (max)	-	425
Sulfur, Wt. percent max.	D1266	0.10
Phosphorous, theory	-	0.0
R.V.P. lb.	D 323	5.5-7.5
Washed gum (max) mgm/gal	D 381	4.0
Corrosion (not lower than)	D 130	1B
Oxidation stability (not less than)	D 525	240+
Hydrocarbon composition	D1319	-
Olefins, percent, max.	-	30
Aromatics, percent, max.	-	40
Saturates	-	Remainder

For computation purposes the lower heating values of this fuel is to be assumed as 18500 Btu/lb. The cost to be assumed for system cost analysis is $0.31/gallon. An A.P.I. gravity of 56.0 is to be assumed in all calculations.

11. START UP, ACCELERATION, AND GRADE VELOCITY PERFORMANCE

a. Startup:

The vehicle must be capable of being tested in accordance with the procedure outlined in the July 2, 1971 Federal Register without special driver startup/warmup procedures. The accessory load shall be as defined in Section 14b.

The maximum time from "key on" to reach 65 percent of full power level is 45 sec. Ambient conditions are 14.7 psia 60°F. The vehicle is to be soaked at this temperature for a minimum of 12 hours prior to initiation of start test.

Powerplant starting procedures in low ambient temperatures shall be equivalent to or better than the typical automobile spark-ignition engine. After a 24 hour soak at -20°F and 14.7 psia the engine shall achieve a self-sustaining idle condition without further driver input within 25 seconds. No starting aids external to the normal vehicle system shall be needed at or above -20°F.

b. Idle operation conditions:

The idle creep torque shall not result in level road operation of the vehicle at a speed in excess of 18 mph in high gear, with the entire propulsion system at steady state operating temperature and ambient conditions of 14.7 psia and 85°F. The accessory load shall be as defined in Section 14a.

c. Acceleration from a standing start:

The minimum distance to be covered in 10.0 sec. is 440 ft. The maximum time to reach a velocity of 60 mph is 13.5 sec. Ambient conditions are 14.7 psia, 85°F. Vehicle weight is W_t and accessory load as defined in Section 14a. Acceleration is on zero grade and initiated with the engine at the normal idel condition.

d. Acceleration in merging traffic:

The maximum time to accelerate from a constant velocity of 25 mph to a velocity of 70 mph is 15.0 sec. Ambient conditions are 14.7 psia, 85°F. Vehicle weight is W_t and accessory load as defined in Section 14a. Acceleration is on zero grade and time starts when the accelerator pedal is depressed.

e. Acceleration, DOT High Speed Pass Maneuver:

The maximum time and distance to go from an initial velocity of 50 mph with the front of the automobile (18 foot length assumed) 100 feet behind the back of a 55 foot truck traveling at a constant 50 mph, to a position where the back of the automobile is 100 feet in front of the front of the 55 foot truck, is 15 sec. and 1400 ft. The entire maneuver takes place in a traffic lane adjacent to the lane in which the truck is operated. Vehicle is accelerated until the maneuver is completed or until a maximum speed of 80 mph is attained, whichever occurs first. Vehicle acceleration ceases when a speed of 80 mph is attained, the manuever then being completed at a constant 80 mph. (This does not imply a design requirement limiting the maximum vehicle speed to 80 mph.) Time starts when the accelerator pedal is depressed. Ambient conditions are 14.7 psia, 85°F. Vehicle weight is W_t and accessory load as defined in Section 14a. Acceleration is on zero grade.

f. Grade velocity:

The vehicle must be capable of starting from rest on a thirty percent (30%) grade and ascending the grade at a minimum speed of 5 mph. The vehicle must be capable of this maneuver in both the forward and reverse directions at a vehicle weight of W_g, with the accessory load as defined in Section 14a.

The minimum cruise velocity that can be continuously maintained on a five percent (5%) grade shall be not less than 65 mph with a vehicle weight of W_g and accessory load as defined in Section 14a.

The minimum cruise velocity that can be continuously maintained on a zero percent (0%) grade shall be not less than 85 mph with a vehicle weight of W_t and with the accessory load as defined in Item 14a.

Ambient conditions for all grade specifications are 14.7 psia and 85°F.

Performance degradation attributable to loss of powerplant efficiency at extreme temperatures shall not exceed ten percent (10%) relative to the performance values specified at 85°F. This limitation applies to ambient temperatures from -20°F to 105°F.

The wind velocity is to be less than 10 mph for all acceleration and grade tests.

12. MINIMUM VEHICLE RANGE

Minimum vehicle range without refueling will be 200 miles (maximum fuel capacity is 25 U.S. gallons). The minimum range shall be calculated for, and applied to each of the following modes:

1.	Cyclic mode is:	The Federal driving cycle which is in accordance with the July 2, 1971 Federal Register. The range may be calculated for one cycle and ratioed to 200 miles.
2.	Cruise mode is:	A constant 70 mph cruise on a zero grade for 200 miles.

The vehicle weight for both modes shall be W_t initially and with accessory power levels as specified in Section 14. The ambient conditions shall be a pressure of 14.7 psia, and a temperature of -20°F (air conditioner off) and 105°F (air conditioner on).

13. FUEL CONSUMPTION

Using the fuel specified in Section 10, a "fuel economy" figure shall be calculated based on 1) miles per gallon and 2) the number of Btu per mile required to drive the vehicle through the following modes of operation:

		Avg. Speed	Hours	% of Time
1)	Federal Driving Cycle	19.84	1750	50
2)	Simplified Surburban Route (equal times at constant 20, 30 and 40 mph speeds).	30.00	1150	33
3)	Simplified Country Route (equal times at constant 50, 60 and 70 mph speeds).	60.00	600	17
	Totals	30	3500	100

In all cases the system fuel consumption shall be calculated for a vehicle weight of W_t initially, and power levels as specified in item 14a. Ambient conditions are 14.7 psia and 85°F.

It is desirable that the fuel consumption rate at idle operating condition not exceed 7 lbs/hour.

14. ACCESSORY POWER REQUIREMENTS

a. Accessory power requirements with the air conditioning in operation are defined as 15 hp at maximum engine speed and 4 hp at engine idle speed, with a linear relationship between these two points.

b. Accessory power requirements without the air conditioning in operation are defined as 5 hp at maximum engine speed and 2 hp at engine idle speed, with a linear relationship between these two points.

15. PROPULSION SYSTEM OPERATING TEMPERATURE AND PRESSURE RANGE

The propulsion system shall be operable within an expected ambient temperature range of -40° to 125°F.

The propulsion system shall be operable within an expected environmental pressure range of 9 psia to 15 psia.

16. PASSENGER COMFORT REQUIREMENTS

Heating and air conditioning of the passenger compartment shall be at a rate equivalent to that provided in the present (1970) standard full size family car.

Present practice for maximum passenger compartment heating rate is approximately 30,000 Btu/hr. For an air conditioning system at 110°F ambient, 80°F and 40% relative humidity air to the evaporator, the rate is approximately 13,000 Btu/hr.

17. NOISE STANDARDS

a. Maximum noise test:*

The maximum noise generated by the vehicle shall not exceed 77 dbA when measured in accordance with SAE J986a. Note that the noise level is 77 dbA whereas in the SAE J986 the level is 86 dbA.

b. Low speed noise test:*

The maximum noise generated by the vehicle shall not exceed 63 dbA when measured in accordance with SAE J986a except that a constant vehicle velocity of 30 mph is used on the pass-by.

c. Idle noise test:*

The maximum noise generated by the vehicle shall not exceed 62 dbA when measured in accordance with SAE J986a except that the engine is idling (clutch disengaged or in neutral gear) and the vehicle is stationary. A 360° survey shall be made, the microphone being 10 feet from the vehicle perimeter.

18. OPERATIONAL LIFE

The design lifetime of the propulsion system in normal operation will be 3500 hours minimum.

Terminationof the operational life of an engine shall be determined by structural or functional failure. Functional failure is defined as power degradation exceeding 25 percent of maximum output of the rear wheels.

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*The air conditioner will not be in operation during noise tests.

19. RELIABILITY AND MAINTAINABILITY

The reliability and maintainability of the vehicle shall equal or exceed that of the spark-ignition automobile. The mean-time-between-failure should be maximized to reduce the number of unscheduled service trips. No failure modes shall present a serious safety hazard during vehicle operation and servicing. Failure propagation should be minimized. The power plant should be designed for ease of maintenance and repairs to minimize costs, maintenance personnel education, and downtime.

20. COST OF OWNERSHIP

The initial cost and net cost of ownership of the vehicle shall be minimized for ten years and 105,000 miles of operation.

21. SAFETY STANDARDS

The vehicle shall comply with all Department of Transportation Federal Motor Vehicle Safety Standards in force when the selected test vehicle was manufactured.

TABLE A-1 (Excerpt)

TEST CONDITIONS
(ENGINE DYNAMOMETER, CHASSIS DYNAMOMETER, ROAD)

Section	Performance Requirements	Accessory Power	Weight	Temperature	Pressure
9. EMISSIONS	HC 0.20 grams per mile**	14a	W_t	85°F	14.7
	CO 1.70 grams per mile**	14a	W_t	85°F	14.7
	NO_2 0.20 grams per mile**	14a	W_t	85°F	14.7
11. PERFORMANCE a. Start Up	65%* power in 45 sec**	14b	W_t	60°F	14.7
	Driver Assistance 25 sec**	14b	W_t	-20°F	14.7
b. Idle	Creep 18 mph**	14a	W_t	85°F	14.7
c. Accel (0-60)	13.5 sec** to 60 MPH	14a	W_t	85°F	14.7
	440 ft* in 10.0 sec	14a	W_t	85°F	14.7
d. Accel (25-70)	15.0 sec** from 25 to 70 MPH	14a	W_t	85°F	14.7
e. Accel (50-80)	15.0 sec** and 1400 Ft** From 50 to 80 MPH**	14a	W_t	85°F	14.7
f. Grade (30%)	0 to 5 MPH*	14a	W_g	85°F	14.7
(5%)	65 MPH*	14a	W_g	85°F	14.7
(0%)	85 MPH*	14a	W_t	85°F	14.7
12. VEHICLE RANGE	200 MI* 1. During FCE	14b and 14a	W_t	-20°F and 105°F	14.7
	2. At 70 MPH	14b and 14a	W_t	-20°F and 105°F	14.7
13. FUEL CONSUM.	MPG During FDC	14a	W_t	85°F	14.7
	MPG at 20, 30, and 40 MPH	14a	W_t	85°F	14.7
	MPG at 50, 60, and 70 MPH	14a	W_t	85°F	14.7

NOTES: Emission tests will be run with fuel specified in Section 10.
Road test wind conditions shall not exceed 10 MPH in any direction.

*Minimum values

**Maximum values

APPENDIX B

ATTACHMENT

ENVIRONMENTAL PROTECTION AGENCY

PRELIMINARY
PROTOTYPE VEHICLE SPECIFICATION
COMPACT PASSENGER CAR

FEBRUARY 1974

Alternative Automotive Power
Systems Division

2929 Plymouth Road
Ann Arbor, Michigan 48105

ALTERNATIVE AUTOMOTIVE POWER SYSTEMS DIVISION (AAPSD)

PROTOTYPE VEHICLE SPECIFICATION COMPACT PASSENGER CAR

This specification defines emissions, fuel economy, noise and performance requirements for automobiles intended to be powered by other than spark ignition engines. Emissions, volume and most weight characteristics presented are maximum values. Performance characteristics are intended as minimum values.

Significant changes over previous specifications are in the vehicle size and performance requirements.

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1. VEHICLE WEIGHT WITHOUT PROPULSION SYSTEM - W_o

W_o is the weight of the vehicle excluding the propulsion system. This weight includes, but is not limited to, the frame, body, glass, trim, suspension, wheels (rims and tires), service brakes, seats, upholstery, sound absorbing materials, insulation, dashboard instruments, accessory ducting and wiring, accessories, and all other components not included as part of the propulsion system.

Accessories are defined as driver assistance and passenger convenience components and subsystems not essential to propulsion system operation. Included are power steering systems, power brake systems and passenger compartment heating and air conditioning systems.

The maximum allowable weight W_o is 2000 lbs.

2. PROPULSION SYSTEM WEIGHT - W_p

W_p includes the fuel storage subsystem (including fuel, containment, and supply and deliver ducting), power conversion subsystem (including auxiliaries and controls) and power transmitting subsystem (including transmission and drive train to the driven wheels).

Auxiliaries are defined as components and subsystems essential to the operation of the propulsion system. Included are electric power generating subsystem, starting subsystem, exhaust subsystem, motors, fans, blowers, pumps, auxiliary drives, and fluids.

The maximum allowable propulsion system weight, W_{pm}, is 1000 lbs.

3. VEHICLE CURB WEIGHT - W_c

$W_c = W_o + W_p$.

The maximum allowable vehicle curb weight, W_{cm}, is 3000 lbs.

4. VEHICLE TEST WEIGHT - W_t

$W_t = W_c + 400$ lbs. W_t is the vehicle weight at which all accelerative maneuvers, fuel economy and emissions are to be calculated. (Items 9, 11c, 11d, 11f, 12, 13.)

The maximum allowable test weight, W_{tm}, is 3400 lbs. For chassis dynamometer testing, maximum inertia weight is 3500 lbs.

5. GROSS VEHICLE WEIGHT - W_g

$W_g = W_c + 700$ lbs. W_g is the gross vehicle weight at which the DOT pass maneuver (11e) and sustained velocity capability at 30% grades is to be calculated. (Item 11f.) The 700 lb load simulates a full load of four passengers and baggage.

The maximum allowable gross vehicle weight, W_{gm}, is 3700 lb.

6. PROPULSION SYSTEM VOLUME - V_p

V_p is the volume allotted for all items identified under item 2. The propulsion system shall be packagable in such a way that the volume encroachment on either the passenger or luggage compartment does not exceed the following:

a) The transmission tunnel may not be widened so as to decrease the selected production vehicle clearance between the accelerator pedal and the tunnel. The accelerator pedal may not be relocated.

b) Intrusion of the tunnel into the passenger side of the vehicle may be increased by a maximum of 1.5 inches.

c) The tunnel height may be increased by a maximum of 2 inches but without affecting the full fore and aft adjustment of the front seat of the vehicle. The front seat may not be raised.

The propulsion system shall not violate the vehicle ground clearance lines as established by the manufacturer of the vehicle used for propulsion system/vehicle packaging. Additionally, the propulsion system shall not violate the space allocated for wheel jounce motions and vehicle steering clearances. Necessary external appearance (styling) changes will be minor in nature. The propulsion system shall also be packagable in such a way that the handling characteristics of the vehicle are not degraded.

7. AIR DRAG

The product of the drag coefficient, C_d, and the frontal area, A_f, to be used in air drag calculations shall be 10 ft^2. The air density used in computations shall correspond to the applicable ambient air temperature.

8. ROLLING RESISTANCE

Rolling resistance, R, is expressed in the equation $R = (W/65) [1 + (1.4 \times 10^{-3}V) + (1.2 \times 10^{-5}V^2)]$ lbs. V is the vehicle velocity in ft/sec. W is the vehicle weight in lbs.

9. PROPULSION SYSTEM EMISSIONS

a. Exhaust Emissions

The vehicle is to be tested for emissions in accordance with the procedure in the November 15, 1972 Federal Register for model year 1975 light duty vehicles. The vehicle test weight shall be as specified in the Federal Register for the actual curb weight W_t. The accessory load shall be as defined in Section 14 without air conditioning in operation. Ambient conditions are 14.7 psia and 85°F. Emissions tests will be run with fuel specified in Section 10.

The applicable Federal emissions standards are:

Hydrocarbons - 0.41 grams/mile maximum
Carbon Monoxide - 3.40 grams/mile maximum
Oxides of Nitrogen* - 0.40 grams/mile maximum

Prototype Vehicles shall meet the following emissions goals when tested as specified above. Measurement of emissions is to be taken after the system has accumulated 100 hours.

Hydrocarbons - 0.20 grams/mile maximum
Carbon Monoxide - 1.70 grams/mile maximum
Oxides of Nitrogen* - 0.20 grams/mile maximum

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*Oxides of nitrogen are to be measured or computed as NO_2.

Production of smoke, odors, aldehydes, ammonia, particulates or other emissions not now specified in the November 15, 1972 Federal Register are undesirable. Tests for smoke shall utilize instrumentation in accordance with paragraph 85.125 of the Federal Register November 10, 1970. Tests for odor shall be by the Quality Intensity (Turk) method.

b. Fuel Evaporative Emissions

Fuel Evaporative Emissions shall be measured in accordance with the procedure in the November 15, 1972 Federal Register. The Fuel Evaporative Emission standard is 2.0 grams per test.

10. FUEL

a. Test Fuel

Emission tests will use the fuel specified below, however, the power system shall have the capability of meeting the applicable Federal emission standards above using other commercially available fuels as stated in paragraph 10b.

Item	ASTM Designation	Specification
Octane, Research, min.	D2699	91-93
Pb (Organic) gm/U.S. gal	D 524	<0.02
Distillation range	D 86-67	-
I.B.P., °F	-	100-115
10 percent point, °F	-	140-150
50 percent point, °F	-	240-250
90 percent point, °F	-	330-340
E.P. °F (max)	-	425
Sulfur, Wt. percent max.	D1266	0.10
Phosphorous, theory		0.0
R.V.P. lb.	D 323	5.5-7.5
Washed gum (max mgm/100 ml)	D 381	4.0
Corrosion (not lower than)	D 130	1B
Oxidation stability (not less than)	D 525	240+
Hydrocarbon Composition	D1319	-
Olefins, percent, max.	-	30
Aromatics, percent, max.	-	40
Saturates	-	Remainder
Nitrogen, Wt % max.**	-	0.005

**Nitrogen content shall include chemically bound plus additive introduced. Measurement shall be by Kjeldahl method.

For computation purposes the lower heating value of this fuel is to be assumed as 18500Btu/lb. An A.P.I. gravity of 56.0 is to be assumed in all calculations except where actual measurements are available.

b. Alternative Fuels

In order to accommodate future requirements for multi-fuel capabilities, new designs should consider presently available transportation fuels such as Jp5, Jp4, Kerosene, unleaded gasoline, Diesel #1 and Diesel #2. Where using fuel other than unleaded gasoline, minor modifications to the fuel and combustion system are acceptable.

11. START UP, ACCELERATION, AND GRADE VELOCITY PERFORMANCE

The wind velocity is to be less than 10 mph for all acceleration and grade tests.

a. Startup

The vehicle must be capable of being tested in accordance with the procedure outlined in the November 15, 1972 Federal Register without special driver startup/warmup procedures. The accessory load shall be as defined in Section 14, without air conditioning in operation.

The start up performance characteristics of the engine shall be demonstrated by achieving a self sustaining idle condition within 30 seconds from "key on", followed by a 10 second period of increasing power equivalent to an acceleration rate, from a standing start, of 3 miles per hour per second for a vehicle weight of W_t on a zero grade. Ambient pressure shall be 14.7 psia. Ambient temperature shall be between 68 and 85°F.

Powerplant starting procedures at low ambient temperatures shall be equivalent to or better than the typical pre-control automobile spark-ignition engine. No starting aids external to the normal vehicle system shall be needed after a 24 hour soak at -20°F.

b. Idle Operation Conditions:

The idle creep torque shall not result in level road operation of the vehicle at a speed in excess of 15 mph in high gear, with the entire propulsion system at steady state operating temperature and ambient conditions of 14.7 psia and 85°F. The accessory load shall be as defined in Section 14, without air conditioning in operation.

c. Acceleration from a Standing Start:

The maximum distance to be covered in 10.0 sec. is 440 ft. The maximum time to reach a velocity of 60 mph is 17.5 sec. Ambient conditions are 14.7 psia, 85°F. Vehicle weight is W_t and accessory load as defined in Section 14 without air conditioning in operation. Acceleration is on zero grade and time starts when the accelerator pedal is depressed.

d. Acceleration in Merging Traffic:

The maximum time to accelerate from a constant velocity of 25 mph to a velocity of 70 mph is 20.0 sec. Ambient conditions are 14.7 psia, 85°F. Vehicle weight is W_t and accessory load as defined in Section 14 without air conditioning in operation. Acceleration is on zero grade and time starts when the accelerator pedal is depressed.

e. Acceleration, DOT High Speed Pass Maneuver:

The maximum time to complete the DOT High Speed Pass Maneuver is 17 seconds. The entire maneuver takes place in a traffic lane adjacent to the lane in which the truck is operated. Vehicle is accelerated until the maneuver is completed or until a maximum speed of 80 mph is attained, whichever occurs first. Vehicle acceleration ceases when a speed of 80 mph is attained, the maneuver then being completed at a constant 80 mph. Time starts when the accelerator pedal is depressed. Ambient conditions are 14.7 psia, 85°F. Vehicle weight is W_g and accessory load as defined in Section 14 without air conditioning in operation. Acceleration is on zero grade.

f. Grade Velocity:

The vehicle must be capable of starting from rest on a thirty percent (30%) grade and ascending the grade at a minimum speed of 5 mph. The vehicle must be capable of this maneuver in both the forward and reverse directions at a vehicle weight of W_g, with the accessory load as defined in Section 14, without air conditioning in operation.

12. MINIMUM VEHICLE RANGE

Minimum vehicle range without refueling will be 300 miles for a constant 50 mph cruise on a zero grade.

The vehicle weight shall be W_g initially and with accessory power levels as specified in Section 14. The ambient conditions shall be a pressure of 14.7 psia, and a temperature of -20°F (air conditioner off) and 105°F (air conditioner on). Maximum fuel capacity shall be 15 U.S. gallons.

13. FUEL ECONOMY

a. Fuel Economy of Current Vehicles

The average fuel economy over the Federal Driving Cycle for 1400 tests on 1973 model year vehicles was reported in SAE paper 730790. Correcting that analysis for the 1975 test procedure gives the following equation for fuel economy as a function of weight:

$$MPG_{(avg)} = \frac{55050}{W_t} - 1.34 \quad (14.9 \text{ mpg for } W_t = 3400 \text{ lb})$$

b. Fuel Economy Goal

The goal for first generation prototype hardware is to equal 130% of the fuel economy shown by the equation. (1.3 x 14.9 = 19.4 mpg for wt = 3400 lb).

In addition to the goal stated above, fuel economy shall be measured for the component parts of the following driving profile, and the equivalent miles per gallon calculated for the overall model.

Mode	Mode Usage	Miles Per Year	Hours Per Year
Federal Driving Cycle	465 23-minute runs 1 Cold start each run	3500	177
30 mph	50 12-minute runs 1 Hot start each run	300	10
40 mph	125 12-minute runs 1 Hot start each run	1000	25
50 mph	150 12-minute runs 1 Hot start each run	1500	30
60 mph	175 12-minute runs 1 Hot start each run	2100	35
70 mph	114 12-minute runs 1 Hot start each run	1600	23
Total	465 Cold starts, 614 Hot starts	10,000 mi.	300 hr.

The goal for idle fuel consumption will not be greater than 5 pounds per hour.

In all cases, fuel economy shall be determined including suitable accessory loads in accordance with Section 14. A separate determination shall be made with and without air conditioning. The goals stated above shall be met without air conditioning. Ambient conditions are 14.7 psia and 85°F. Cold start means the engine has been soaked at the ambient condition for at least 12 hours. Hot start means the engine is started with no greater than 15 minutes elapsed from the prior complete shut down.

For prototype hardware, a complete map of BSFC shall be obtained, with fuel consumption referenced to the horsepower at the transmission input coupling, and all engine auxiliaries deriving power from the engine output torque. (No Test Support Equipment.)

14. ACCESSORY POWER REQUIREMENTS

In all cases accessory power consumption shall be minimized. Consideration shall be given to accessory power subsystems which are de-coupled from the torque-speed-power characteristics of the engine. In the absence of other experimental data, the following equation shall be used to determine accessory power. Without air conditioning in operation:

$$hp = 1.5 + (.333 \times 10^{-3})(rpm)$$

With air conditioning in operation:

$$hp = 3.0 + (1.83 \times 10^{-3})(rpm)$$

where hp = accessory power
rpm = shaft speed of the accessory drive shaft

15. PROPULSION SYSTEM OPERATING TEMPERATURE AND PRESSURE RANGE

The propulsion system shall be operable within an expected ambient temperature range of -40° to 125°F.

The propulsion system shall be operable within an expected ambient pressure range of 9 psia to 15 psia.

Performance degradation attributable to loss of powerplant efficiency at extreme temperature shall not exceed ten percent (10%) relative to the performance values specified at 85°F. This limitation applies to ambient temperatures from -20°F to 105°F.

16. PASSENGER COMFORT REQUIREMENTS

Heating and air conditioning of the passenger compartment shall be at a rate equivalent to that provided in the present compact size passenger car.

Present practice for maximum passenger compartment heating rate is approximately 20,000 Btu/hr. For an air conditioning system at 110°F ambient, 80°F and 40% relative humidity air to the evaporator, the rate is approximately 10,000 Btu/hr.

17. NOISE STANDARDS

a. Maximum Noise

The maximum noise generated by the vehicle shall not exceed 77 dbA when measured in accordance with SAE J986a. Note that the noise level is 77 dbA whereas in the SAE J986 the level is 86 dbA.

b. Low Speed Noise

The maximum noise generated by the vehicle shall not exceed 63 dbA when measured in accordance with SAE J986a except that a constant vehicle velocity of 30 mph is used on the pass-by.

c. Idle Noise

The maximum noise generated by the vehicle shall not exceed 62 dbA when measured in accordance with SAE J986a except that the engine is idling (clutch disengaged or in neutral gear) and the vehicle is stationary. A 360° survey shall be made, the microphone being 10 feet from the vehicle perimeter.

*The air conditioner will not be in operation during noise tests.

18. OPERATIONAL LIFE

The design lifetime of the propulsion system in normal operation will be 3500 hours minimum.

Termination of the operational life of an engine shall be determined by structural or functional failure. Functional failure is defined as power degradation exceeding 25 percent of maximum output of the rear wheels.

19. RELIABILITY AND MAINTAINABILITY

The reliability and maintainability of the vehicle shall equal or exceed that of the spark-ignition engine automobile. The mean-time-between-failure should be maximized to reduce the number of unscheduled service trips. No failure modes shall present a serious safety hazard during vehicle operation and servicing. Failure propagation should be minimized. The power plant should be designed for ease of maintenance and repairs to minimize costs, maintenance personnel education, and downtime.

20. COST OF OWNERSHIP

The initial cost and net cost of ownership of the vehicle shall be minimized for ten years and 105,000 miles of operation.

21. SAFETY STANDARDS

The vehicle shall comply with all Department of Transportation Federal Motor Vehicle Safety Standards in force when the selected test vehicle was manufactured.

22. VEHICLE HANDLING

The use of an alternative engine shall not significantly change the handling characteristics ("feel") of the vehicle. To demonstrate this, the vehicle handling characteristics of alternative engine powered vehicles will be compared to conventionally powered vehicles with as close to the same overall configuration as possible. The handling performance goals shall be:

a. Low Speed Maneuverability (Slalom)

The maximum safe speed at which the vehicle can maneuver through pylons will be determined. Pylon spacing will be 48 feet, with 12 feet wide lanes. Maximum test speed will be 25 mph.

b. Braking Distance

The braking distance will be measured from 30 mph to stop within a 12 foot wide lane.

c. Cornering

The maneuverability of the vehicle into and around both 50 and 100 foot diameter circles will be determined. Maximum test speed on the 50 foot circle will be 15 mph; and maximum test speed on the 100 foot circle will be 22 mph.

d. Lane Change (Swerve)

The maximum safe speed for changing from one 12 foot wide lane to another 12 foot wide lane and back over a total distance of 240 feet will be determined. The maximum test speed will be 50 mph.

e. Smoothness (Driveability)

The smoothness of the engine-vehicle operation at idle and between 25 and 35 mph will be determined, including the relative idle roughness, hesitation, tip-in, stumble, surge, detonaton, backfire, stall and after-run (dieseling) as applicable.

ABBREVIATIONS

AAPS	Alternative Automotive Power Systems
ac	alternating current
Ah	ampere hours
Ah/cm^2	ampere hours per square centimeter
ALRC	Aerojet Liquid Rocket Company
AMC	Army Materiel Command
AMMRC	Army Materials and Mechanics Research Center
APCO	Air Pollution Control Office (EPA), formerly NAPCA
API	a method (American Petroleum Institute) of expressing specific gravity of hydrocarbons: $\text{API} = \frac{141.5}{\text{specific gravity } 60°\text{F}/60°\text{F}} - 131.5$
APL	Applied Physics Laboratory, John Hopkins University
ARB	Air Resources Board (California)
ARPA	Advanced Research Projects Agency (DDR&E)
ASME	American Society of Mechanical Engineers
ASTM	American Society for Testing Materials
BDC	bottom dead center
bhp	brake horse power
BSFC	brake specific fuel consumption
BMEP	brake mean effective pressure
Btu	British thermal unit
C_2H_5-OH	chemical formula for ethanol (same as ethyl alcohol)
CCCP	California Clean Car Project
CECO	Chandler Evans Control Systems
CFR	Cooperative Fuel Research (Council)
CH_4	chemical formula for methane

CH_3OH	chemical formula for methyl alcohol (methanol)
CNG	compressed natural gas
CO	chemical formula for carbon monoxide
CO_2	chemical formula for carbon dioxide
CI	compression ignition
CID	cubic inches of displacement
CVCC	Compound Vortex Controlled Combustion (Honda)
CVS	constant volume sampling
dB(A)	decibel referred to one ampere
dc	direct current
DCF	Discounted Cash Flow, a method of computing return on investment
DOT	Department of Transportation
EFC	electronic fuel control
EFE	early fuel injection
EGR	exhaust gas recirculation
EMT	electromechanical transmission
EPA	Environmental Protection Agency (formerly HEW)
EVC	external vaporizing combustion
FCP	Ford Combustion Process
FDC	Federal Driving Cycle(test for determining exhaust emissions)
FSTP	Federal Smoke Test Procedure
FTP	Federal Test Procedure
g/mi	grams per mile (also gm/mi)
gvm	grams per vehicle mile
H_2	chemical formula for hydrogen

HC	hydrocarbon
HEW	Department of Health, Education, and Welfare (predecessor to EPA in studying the control of automotive emissions)
hp	horsepower
hp-hr	horsepower per hour
ICE	internal combustion engine
ihp	indicated horsepower
IMEP	indicated mean effective pressure
ISFC	indicated specific fuel consumption
JIC	jet-induced circulation
KKM-125	Kreiskolbenmotor, orbiting piston engine, 125 cubic centimeters in volume
kW	kilowatt
LAS	lithium-alumina-silicate
lb	pound
lb/ihp-hr	pounds per indicated horsepower hour
LMC	Lear Motors Corporation
LPG	liquified petroleum gas
MCP	Mitsubishi Combustion Process
MON	motor octane number
mpg	miles per gallon
MTI	Mechanical Technology, Inc.
NAPCA	National Air Pollution Control Administration (see APCO)
NH_3	chemical formula for ammonia
N_2	chemical formula for nitrogen
N_2H_4	chemical formula for hydrozene

NO_x	chemical formula for oxides of nitrogen
O_2	chemical formula for oxygen
PCU	power control unit
PWM	pulse-width modulation
PROCO	Programmed Combustion (Ford)
psi	pound per square inch
psia	pound per square inch, absolute
psig	pound per square inch, gage
PTX	Engelhard PTX, oxidation catalyst
rms	root mean square
RON	research octane number
rpm	revolutions per minute
SCE	stratified charge engine
SCFM	standard cubic feet per minute flow of gas, measured at $70^{\circ}F$ and 14.696 psia
SCR	silicon controlled rectifier
SCRTD	Southern California Rapid Transit District
SES	Scientifc Energy Systems
SFC	specific fuel consumption
SI	spark ignition
Si_3N_4	silicon nitride
SNG	synthetic natural gas
SPS	Steam Power System
SWRI	Southwest Research Institute
TACOM	(U.S. Army) Tank Automotive Command

TCCS	Texaco Controlled Combustion System
TC-25	transit coach
TCP	Texaco Combustion Process
TDC	top dead center
TECO	Thermo Electron Company
UACRL	United Aircraft Corp. Research Laboratory
USATACOM	(see TACOM)
V	volt
V_p	propulsion system volume (see Appendix A)
W	watt
Wh	watt hour
Wh/lb	watt hours per pound
WOT	wide-open throttle

GLOSSARY

aldehydes	an odorous, oxygenated hydrocarbon classified as a pollutant
Anhydrous	without water
aspiration (British)	process by which engine takes in air fuel mixture (breathing)
autoignition	spontaneous ignition
base metal	a metal or alloy (as zinc, lead, or brass) of comparatively low value and relatively inferior in certain properties (as resistance to corrosion)--opposed to noble metal
brake mean effective pressure (BMEP)	the effective constant pressure required to be exerted during each stroke to produce the power delivered by the engine
brake horsepower	the net power available at the output shaft of the engine
brake specific fuel consumption	the fuel flow rate required per unit of net horsepower output of the engine
Brayton cycle	a thermodynamic cycle, consisting of gas compression heating and expansion stages without phase changes of gas; the operating basis of a gas turbine engine
catalyst	a substance that increases the rate of a chemical reaction without itself undergoing a permanent chemical change
catalytic converter	as applied to automotive heat engines, a device relying on a catalyst in the engine exhaust with the objective of lessening pollutant emissions to the atmosphere
cetane number	indicates the ignition delay of a fuel in a compression ignition (diesel) engine
displacement	the volume swept out by the pistons or other power elements

endothermic reaction	chemical reaction in which heat is absorbed
equivalence ratio	the actual fuel air ratio to stoichiometric fuel air ratio
exothermic reaction	a chemical reaction in which heat is evolved
flash point	the temperature at which the bulk liquid can be ignited by an open flame
gross or higher heating value	the energy released by a fuel when it is completely burned in such a way that the vapor of combustion is condensed to a liquid
hypergolic ignition	spontaneous ignition upon contact of the fuel and oxidizer
investment casting	a process used in metal casting that consists of making a wax model, coating it with a refractory to form a mold, heating until the wax melts, and then pouring metal into the space left vacant
kinematic viscosity	viscosity divided by density
naphtha	a light oil product of petroleum having properties intermediate between gasoline and kerosene
natural gas	primarily a gaseous fuel consisting of methane
neat	a pure material without additives
net or lower heating value	the energy released by a fuel when it is burned completely without condensing the water vapor of combustion
octane number	a measure of the resistance of a fuel to detonate in a spark-ignition, reciprocating engine (Otto cycle engine)
Rankine cycle engine	an external combustion engine in which a high-pressure working fluid is converted to vapor by the heat from combustion gases and then is expanded in a piston or turbine expander to produce work.
recuperator	a fixed device used to improve gas turbine engine efficiency by providing for heat exchange between turbine exhaust gases and compressor outlet air

regenerator	a rotating device used to improve gas turbine engine efficiency by providing for heat exchange between turbine exhaust gases and compressor outlet air
regenerative engine	incorporates a regenerator or recuperator to improve engine efficiency
sink	a body or substance used for the disposal of a fluid or heat in the course of a hydrodynamic or a thermodynamic process (as the condenser of a steam engine)
Stirling cycle engine	an external combustion, closed cycle, piston-type device that uses a gaseous internal working fluid, usually hydrogen or helium. Cyclical heating and cooling varies the pressure of the fluid within a closed volume, the pressure variations being transmitted to a piston, thereby developing output power.
stoichiometric	refers to conditions in a combustion process wherein the fuel is completely burned with no excess oxygen remaining
syncrude	a synthetic crude oil made from oil shale or coal
thermal efficiency	the ratio of theoretical work done by an engine to the mechanical equivalent of heat supplied in the fuel
thermal reforming	a process using heat (but no catalyst) to effect molecular rearrangement of a hydrocarbon fuel (example: low-octane naphtha into gasoline of higher antiknock quality)
torque	any turning movement exerted by a force acting tangentially at a distance from the axis of rotation, the product of a force and the distance of its line of action from the axis

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